TAMPEREEN TEKNILLINEN YLIOPISTO
TAMPERE UNIVERSITY OF TECHNOLOGY

MIKKO JÄRVINEN

DEVELOPMENT OF ICE CLASS CALCULATION TOOL AND PROCESS SENSITIVITY ANALYSIS

Master of Science thesis

Examiner: prof. Reijo Kouhia
Examiner and topic approved by the Faculty Council of the Faculty of Mechanical Engineering and Industrial Systems
on 8th April 2015

ABSTRACT

MIKKO JÄRVINEN: Development of Ice Class Calculation Tool and Sensitivity Analysis
Tampere University of technology
Master of Science Thesis, 82 pages, 5 Appendix pages
August 2015
Master's Degree Programme in Mechanical Engineering
Major: Analysis of Machines and Structures
Examiner: Professor Reijo Kouhia

Keywords: ice class rules, propulsion, azimuth thruster, power transmission, gear calculation, simulation, torsional vibration

The main focus of the study is in transient torque vibration due to ice excitation on propulsion machinery components but some background for ice class rules is also presented generally. It is studied how the ice classification calculation process for propeller units could be done more automatically and how sensitive the process is to different parameters. Additionally aspects of the upcoming rule updates are discussed. These updates try to harmonize the ice class calculations internationally and bring out alternatives for the calculation process.

As a part of this study partly automated calculation process was developed with Matlab as a sales team tool to ease the preliminary calculations that have to be made during the sales offer. The program includes simulation of a thruster unit and gear calculations based on the given input parameters. This kind of a tool is needed because the offered thruster needs to be classified accordingly and on the current rules this is the safest way to ensure that the offered values can be achieved.

The program was developed for a single thruster unit using Finnish-Swedish ice class rules and marine gear rating rules. First in the program ice class excitation loads are calculated and simulated. Then the gears are evaluated to find out validity of power for the chosen thruster unit. This program can be applied to other units as well by making the necessary changes to the propeller unit's simulation model and by changing the mass-elastic data.

Sensitivity analysis was also conducted for Rolls-Royce US 305 azimuthing thruster unit. The results showed that the most critical factor is the gear design itself because it needs to be optimized for heavy loads and the current designs are based on the free running conditions. Other design factors influence the resulting maximum power as well and may have significance cumulatively but otherwise their effect is minimal compared to the gears. Flexible coupling is also quite critical component because of the damping properties and its effect on systems nominal frequencies.

Higher power can be achieved by choosing a lower inertia engine or motor and choosing a lower input speed. Additionally propeller design parameters like pitch have some effects but the most important factor is the choice between open and ducted propeller. Open propellers have notably higher loads than ducted in the rules. Other important factor in the design is the propeller diameter, bigger diameter increases the ice load.

TIIVISTELMÄ

MIKKO JÄRVINEN: Jääluokkalaskentatyökalun kehittäminen ja laskentaprosessin herkkyysanalyysi
Tampereen teknillinen yliopisto
Diplomityö, 82 sivua, 5 liitesivua
Elokuu 2015
Konetekniikan diplomi-insinöörin tutkinto-ohjelma
Pääaine: Koneiden ja rakenteiden analysointi
Tarkastaja: professori Reijo Kouhia

Avainsanat: jääsäännöt, propulsio, potkurilaite, voimaansiirto, hammaspyörälaskenta, simulointi, vääntövärähtely

Pääpaino tässä työssä on momenttijääherätteen tarkastelussa potkurilaitteistossa, jossa jääheräte aiheuttaa voimansiirtokomponentteihin transientin vääntömomenttikuorman. Tätä kuormaa käytetään komponenttien mitoituksessa ja tässä työssä koko prosessia automatisoidaan. Lisäksi esitetään taustaa jäälaskennalle ja tutkitaan prosessin herkkyyttä eri suunnitteluparametreille. Lopuksi käydään läpi tulevia sääntöpäivityksiä sekä niiden vaikutusta potkurilaitteen jääluokituslaskentaan tulevaisuudessa.

Työn pääkohtana on ruoripotkurilaitteen tarjousvaiheessa tapahtuva laskenta, jonka perusteella tarjottavalle laitteelle lasketaan saavutettavissa oleva teho. Laskentaprosessi on tehtävä, jotta voidaan varmistua siitä, että kaikki komponentit on mahdollista luokittaa hyväksytysti. Kriittisimmät komponentit voimansiirron kannalta ovat hammaspyörät, joiden laskenta sisältyy kehitettyyn Matlab-laskentaohjelmaan. Ohjelma sisältää jäälaskennan suomalais-ruotsalaisten jääsääntöjen mukaisesti, laitteen simuloinnin sekä hammaspyörälaskennan merialuksille tarkoitettujen hammaspyöräsääntöjen mukaisesti. Tuloksena saadaan ovatko hammaspyörät hyväksytyissä rajoissa halutulla teholla.

Ohjelmassa valitaan alkuparametrit kuten jääluokka, potkuritiedot ja laitteen nimellinen maksimiteho. Näiden perusteella ohjelmassa lasketaan ensin jääherätteen suuruus sekä jääiskujen määrä automatisoidusti ja tehdään sen perusteella simulaatio. Simulaatiosta saadaan komponenttien maksimikuormat, joita käytetään hammaspyörälaskennassa. Ohjelma on suunniteltu käytettäväksi yhdelle potkurilaitteelle, mutta muuttamalla laitteen pistemassakaaviota, se on muunnettavissa myös muille potkurilaitteille.

Lisäksi työssä tehtiin herkkyysanalyysiä, jossa tutkittiin tarkemmin Rolls-Roycen US 305 azimuth-potkurilaitetta. Siinä päädyttiin tulokseen, jossa hammaspyöräsuunnittelu on kaikkein kriittisimmässä osassa jääkuormien huomioon ottamisessa. Muillakin komponenteilla on vaikutusta sallittuun maksimitehoon, mutta niiden vaikutus on pieni verrattuna heikkotasoisempaan hammaspyöräpariin. Myös joustavan kytkimen valinta on kriittinen systeemin ominaistaajuuksien ja vaimennuksen kannalta.

Suurempi teho saavutetaan kasvattamalla potkurin hitausmassaa ja vastaavasti vähentämällä sitä moottorilla. Myös teholähteen valinta mahdollisimman pienen kierrosnopeuden mukaan suurentaa tehoa. Näiden lisäksi saavutettavaan tehoon vaikuttavat potkurin suunnittelusuureet kuten nousu, mutta merkittävintä on, että avopotkurille annetaan säännöissä huomattavasti suurempi kuormitus kuin suulakkeelliselle potkurille. Lisäksi potkurin suurempi halkaisija aiheuttaa suuremman kuormituksen.

PREFACE

This thesis was written as a part of Master of Science Degree in Mechanical Engineering at Tampere University of Technology, Finland. The work in this thesis was carried out at Rolls-Royce, Rauma, Finland.

I would like to thank Rami Vanninen and Joni Keski-Rahkonen because without their help and feedback this thesis wouldn’t have been possible. Thanks to other people at Rolls-Royce who helped and gave feedback as well.

Special thanks to my family and friends for all the support.

Rauma, 25.8.2015

Mikko Järvinen

CONTENTS

LIST OF FIGURES

LIST OF TABLES

LIST OF SYMBOLS AND ABBREVIATIONS

C	damping matrix
C_q	torque coefficient [-]
D	propeller diameter [m]
d	propeller hub outer diameter [m]
D_{limit}	limit value for propeller diameter [m]
h_0	depth of the propeller centerline at lower ice water line
H_{ice}	ice design thickness [m]
J	inertia matrix
K	torsional stiffness matrix
K_A	application factor [-]
n	propeller nominal rotational speed [rps]
N_{class}	reference number of impacts per propeller rotational speed per ice class [-]
N_{ice}	total number of ice loads on propeller blade for the ship's service life [-]
$p_{0.7}$	propeller pitch at 0.7R radius [m]
T	torque
T_{max}	maximum torque on the propeller resulting from propeller/ice interaction [kNm]
Z	number of propeller blades [-]
α_i	duration of propeller blade/ice interaction expressed in rotation angle [º]
φ	displacement vector

CP	controllable pitch propeller
CRP	contra-rotating propellers
DIN	Deutsches Institut für Normung, standardization organization
DNV	Det Norske Veritas, classification society
ducted propeller	propeller equipped with a nozzle
FP	fixed pitch propeller
ISO	International Organization for Standardization
load bin	discrete load column
lumped mass	the total mass of the element is distributed to the node
MCR	maximum continuous rating
open propeller	propeller that has its blades fully exposed to the surrounding water
rpm	rotations per minute
scuffing	localized damage caused by welding between sliding surfaces
subsurface fatigue	fatigue damage initiated below the gear tooth surface on surface hardened gears
TraFi	Transport Safety Agency
TVA	torsional vibration analysis

1. INTRODUCTION

Ice classification is an essential way to ensure safety in winter navigation. It determines ice loads that the operating vessels may encounter and requires that the ship's hull, propulsion machinery and other machinery are able to withstand those loads. There exist several different kinds of classification rules which have been developed since the ice navigation has started and the development is still ongoing. Current rules have requirements depending on the vessels operational area and profile.

In this thesis Finnish-Swedish ice classes, which are meant for Baltic Sea area, are studied based on Det Norske Veritas (DNV) classification society rules. Those rules are meant for merchant vessels winter time operations part time of the year but there also exists specific rules for polar area vessels. Some general background is told about these rules and regulations but the main focus of this study is on the propulsion machinery, specifically ice load induced torsional vibration.

Currently most problematic thing about the rules from the azimuthing thruster manufacturer stand point is the requirement for the simulation of the propeller loads affecting the power transmission. This is especially hard considering sales because it basically means that the whole propulsion machinery needs to be designed before actual sales offer can be done. If it is done based on some old projects there are still a lot of uncertainties in the propeller power transmission designs. These uncertainties affect the outcome of allowed power for the ice classified thruster units and those can be really problematic at the later stages of the design.

The power restriction comes mainly from the gear calculation because the gears are only allowed to transmit certain amounts of torque and in ice navigation there are additionally ice induced torque loads present which increase the experienced total torque. The total maximum load comes from the simulation and is applied to the gear calculation. In this study, sales department tool is developed with Matlab so that it would be easier to make these preliminary calculations for sales offers and so that all of the components can be classified in the final design. The tool includes thruster unit simulation and gear calculation.

Also the uncertain factors in the system are studied with sensitivity analysis so that those could be better taken into account already in the sales calculations. This is also important because it greatly affects how competitive choice the offered thruster is compared to the other manufacturers. These uncertainties are mostly caused by different propeller designs and external components.

2. ICE CLASSIFICATION

Depending on the ships operational area and type, it may need to be operated in icy seas at least part time of the year. This is the case at least on the Baltic Sea, Gulf of Finland, Gulf of Botnica and Arctic waters. In order to operate safely and efficiently in these kinds of places, ice loads need to be taken into account and the designed structures have to be able to withstand them. The way it is taken into account is regulated by ice class rules and it needs to be chosen on the vessels design phase based on classification society ice class descriptions and desired ice operations. Vessels operational profile and area are really important factors.

Some of these places are covered in ice only a part of the year and some have multiyear ice and therefore ice strength and other properties are different. In addition encountered ice types are very different. And as is the case in Baltic Sea, vessels operate on open water conditions over half of the year and even on winter there may only be ice on the coastal areas. This makes designing more challenging because ships operation needs to be optimized in different conditions with variable loads.

Therefore accurate description of ships operations and usage is vital for ship design. It affects ship’s hull design, machinery and equipment. Furthermore designer has to understand the effects of the design choices on ice interaction with the help of ice class rules. The rules can’t take into account every possible choice and situation so a good understanding of the experienced loads is vital for the ship and propulsion design.

2.1 Background for ice classification

In Finland, the first ice class rule was issued in 1890. In the beginning the idea was to ensure safer winter navigation but later came along icebreaker fees that are based on ice classes (Riska and Kämäräinen 2011, p. 5). The thought is that vessels which are better equipped for ice conditions have to pay less fairway dues. This is used to encourage getting higher ice class because there are only a limited number of icebreakers available and those need to be capable of assisting every vessel in a reasonable amount of time. The amount of icebreakers has also set the basis for some of the requirements of the Finnish-Swedish ice class rules.

The basic idea is that the highest ice class has no restrictions but the lower ice classes have restricted access as the ice conditions get more severe during the winter (Eriksson et al. 2007, p. 2-3). This can be seen as restrictions to Finnish ports along with the minimum deadweights of the vessels. These restrictions are also used to minimize ice damage to ships as soon as they enter ice (Eriksson et al. 2007, p. 5). Ice classes are also essential in Finland because all of its ports may be ice bound during the winter so in order to maintain economical activity throughout the year, safe and efficient winter navigation system is essential. (Riska and Kämäräinen 2011, p. 2)

Other principles and regulations have affected the rules in other places, for example in Artic waters, but the environmental differences govern the most. In Baltic Sea there is only first year ice part of the year and it doesn’t have swelling like in the oceans. In Arctic seas one can encounter multiyear ice inclusions and the temperatures can be colder. The ice inclusions may also travel more with the winds compared to the Baltic Sea. These kinds of environmental variances mean that ice properties are very different as well. In the Baltic Sea the ice properties are close to constant throughout the winter time (Riska and Kämäräinen 2011, p. 8) which makes certain things easier to assess.

There is constant need for the measurement and research in this field because of the new propulsion or hull solutions in the market. Also there are still unknown aspects how certain factors behave during the propeller-ice interaction for example. There is also all the time increasing demands on pollution and exhaust gas prevention which also will require much work to be done in the future. Pollution prevention is especially important in the arctic area because of the delicate ecosystems.

There have been many updates to ice class rules along the way based on measurements and experience gained in ice navigation but also due to improved computational methods and tools. One big step was the development of the finite element method which provided better and more accurate results. Before that rules were largely based on experience and the need for updates started to rise because the larger ice classes started to be economically inefficient. Other big factor was that Finland and Sweden decided to start developing the rules together as well as organizing the icebreaking services together (Riska and Kämäräinen 2011, p. 6). But even after that there have been many updates and there are also plans for upcoming updates. The planned updates are trying to set the ruling much more uniform globally.

Currently the rules in Northern Baltic are balanced so that the ice capable ships should be able to compete with the ships designed for open water during the summer. The system in Finland uses icebreakers to assist these ships so that they do not need the strengthening as much and can still compete (Riska and Kämäräinen 2011, p. 3). This is also incorporated into the rules by making some requirements about being able to follow icebreakers at brash ice channels. Other requirements mainly depend on the chosen ice class and the operational area.

2.2 Ice classes

Ice class requirements are in addition to open water requirements for the ships. There are several different notations for ice classes and in Table 1 there are the Finnish-Swedish ice classes (left hand side) and the polar classes (right hand side). Finnish-Swedish ice classes are also known as Baltic ice classes and their purpose is to provide design guidelines to merchant ships in part of the year ice covered waters. Polar classes govern more about ships that are meant to sail on arctic or multi-year ice covered waters.

Table 1. *Finnish-Swedish ice classes and polar ice classes.*

Finnish-Swedish ice class	Polar class
	PC-1
	PC-2
	PC-3
	PC-4
	PC-5
IA super	PC-6
IA	PC-7
IB	
IC	
II	
III	

Ice classes of IC and above are the actual ice classes that require ice strengthening and the ice classes II and III are mainly there for fairway fee purposes. IA super is for the most part equivalent to PC-6 and IA to PC-7 so the equivalency from polar class to Baltic class can be admitted by authorities if the power requirements are met. The difference is that the polar classes have more additional requirements for structural strengthening. Finnish-Swedish ice classes are adopted by most of the International Association of Classification Societies members (Trafi 2011, p. 4) but additionally to the presented classes, Russian Maritime Register of Shipping has their own rules for ice classes for the Baltic Sea.

Classification societies also have different kinds of notations for winterization which provide additional measures for vessels operating in low temperatures. These notations rule how de-icing, anti-icing and anti-freezing is done and their goal is to protect shipboard operations and provide personnel welfare. (DNV 2014, p. 79) These kinds of notations are used in addition to ice classes which in turn are in addition to open water requirements.

Ice classes rule what kinds of requirements are set for ship's hull, propulsion and steering machinery in ice navigation and are additional requirements to open water. There are differences in formulations and requirements between Finnish-Swedish and polar classes but concerning this study, the main idea is the same for both types of classifications.

2.3 Ice class requirements

Ice classes require certain things for ships navigating in ice to ensure safety and smooth winter navigation. In this study Finnish-Swedish ice class rules are examined more carefully as these are used later on. Similar requirements can be found for the polar classes as well because the basic principles are the same. The goal is to make the navigation safer and more efficient.

Because there is only a limited amount of icebreakers in Finland, some requirements are set in order to make sure that assistance can be provided for every ship in a reasonable amount of time. This is why there is a minimum speed limit set by the rules so that all the ice classified ships are capable of following icebreakers in ice channels. (Trafi 2011, p. 6) Engine power requirements are based on long term experience and the minimum is set for the same reason (Trafi 2011, p. 6-7). Another benefit of having a minimum requirement on engine output is that it doesn’t favor smaller ships which may have more trouble in ice conditions.

The Finnish-Swedish ice classes do not set any general requirements for hull form. The requirements only take note on hull strengthening like plating, frames and stringers. The basic idea is to strengthen all the parts of the ship that are in close contact with the ice or are exposed to ice loads. This is not only restricted to hull but the propulsion machinery has to be strengthened as well.

The rules set requirements on the propulsion units which are partly examined later on. The main focus on this study is on the power transmission but the rules govern also the amount of experienced load cycles, used materials, propeller blade fatigue stresses on different load cases and required thrust.

One of the key requirements is the time domain simulation of the propeller shaftline because the propeller experiences high ice induced torque loads during propeller-ice interaction. This leads to higher experienced loads on all of the power transmission components and the simulation results are used in the component design. This calculation procedure is therefore basically required already on the sales offer stage of the propeller unit because some of the components restrict the output power greatly.

There are also requirements in the Finnish-Swedish ice classes for steering and rudder arrangements, cooling water etc. And they are to be designed by the rules of chosen classification society as do all the other additional requirements. Classification societies may also have in addition their own rules concerning other machinery or equipment that has to be taken into account. Generally, all the materials need to be chosen and components designed properly for the low temperatures and the possible ice interaction has to be considered.

In polar classes, DNV also has additional requirements for strengthening azimuthing or podded propeller structures. The affecting force on structure is calculated from ice pressure and exposed area with some additional factors for ships operational status and propulsion. Each class has their own value for ice pressure. (DNV 2014, p. 51-53) The resulting forces are then used in evaluation of the propeller unit structures to make sure they can withstand the ice loads.

3. PROPULSION MACHINERY

In this chapter different design factors regarding propulsion machinery are presented. These factors are mainly thought from a perspective of an azimuthing thruster design and are basically design variables that can differ from unit to unit. Azimuthing thrusters differ from more traditional screw propeller-rudder systems by being able to turn 360º around their vertical axis. This means that rudders are not necessary as the maneuverability can be achieved by rotating the propeller unit itself. This also means that the vessel's maneuverability and the flexibility in the machinery arrangement are better.

Just like traditional screw propellers, azimuthing thrusters can be driven with a number of different main machinery and propeller types. The key difference is that the thrusters are a bit more complex solutions as they may require different shaft configurations and more different components.

3.1 Main machinery

Ships main power source can be diesel-electric, geared diesel, gas/steam turbine or nuclear power. This means that the actual propulsion unit is powered by diesel engine, gas turbine or electric motor. Choice of the main machinery or prime mover is governed by vessel arrangement and usage, financial reasons and characteristics it provides. Most common of these drives are geared diesel and diesel-electric which are the main focuses of this thesis.

Diesel engine propulsion is most common in merchant vessels (Trafi 2011, p.10) because it is cheaper to install and maintain. The negative side of that is that the engine has a very limited rpm range where it provides full torque. This means that when ice impacts take place, the engine rpm drops and with high ice torques even stalls. If the diesel engine stalls, it means that the propeller stops rotating and is dragged through ice. If ice impacts happen during this drag, this may cause blade bending damage and that kind of loading condition is not considered as a design criteria and is usually not taken into account.

A favorable choice for propulsion in ice navigation is electric motor system because electric motors are likely to have wider rpm ranges for providing full torque than diesel engines (Riska 2011, p. 14). Other feature that favors electric motors is a possibility to allow brief periods of over torque as sudden and high loads occur on propeller. For example with the ice loads, the motor can maintain the intended speed much better without significant speed drop. This provides much smoother thrust performance (Riska

2011, p. 14-15) but the over torque typically heats up the motor. This requires safety measures to limit the amount of over torque and overheating so that the degradation of motor performance or motor failure does not occur (Tong 2014, p. 7). Electric motor driven propulsion just tends to be more expensive than diesel or gas turbine driven unit and is therefore more common in icebreakers than in merchant vessels (Riska 2011, p. 19).

Depending of the choice of propulsion engine, output speed and power may vary within one propeller unit. This means that the characteristics of the power units differ and several different gear ratios are needed as the propeller speed is wanted to be kept on an optimum range. Azimuthing thrusters also require different shaft line configurations depending on the choice of the power unit. This makes the calculations, general design and comparison of the same type of units more complex.

3.2 Shafting configuration

Propulsion machinery shaft configuration depends a lot from the ship design and the chosen type of engine and propeller. Three typical cases are direct shaft, L- and Z-configuration. Direct shaft is a choice used with traditional propulsion designs but azimuthing propeller units require also other choices. Propeller shaft is in all cases set horizontally but in L-drive input shaft is vertical and on Z-drive it is horizontal. All of these are geared but with direct shaft it may not be necessary if a slow speed engine is used.

With direct shaft, engine is directly connected to propeller shaft and it is possible to use more than one engine with a gearbox. These kinds of drives are common with traditional screw propeller and rudder systems. Special case of direct drive is a podded drive which has electric motor connected directly to propeller shaft inside the propulsion unit. This type of drive can provide better mechanical efficiency and flexibility in general machinery and vessel arrangement. Similar flexibility can also be achieved with other types of azimuthing thrusters.

Typically Z-drive is needed with azimuth thrusters that are driven with diesel engine or gas turbine which is placed inside the ship’s hull. This type of configuration consists of two pairs of bevel gears and three shafts: horizontal input shaft, vertical shaft and horizontal propeller shaft. Although this is mechanically much more complex system it is still useful because of better the maneuverability characteristics. This kind of drive is studied more later on.

L-drive is a modification of Z-drive where there is only one pair of bevel gears and two shafts, horizontal propeller shaft and vertical input shaft. The input shaft is driven by electric motor placed in an upright position on the ship’s hull. The L- and Z-drive both allow more flexible placement of the thruster unit.

Traditionally propellers are placed at the stern with the engines but azimuth thrusters make it possible to locate the propellers at the bow as well. Especially if the chosen power unit for propulsion is electric motor. This may be helpful in managing the available space but this of course has to be taken into account in the design. Bow propellers are much more exposed to possible propeller-ice interaction.

3.3 Components

In the previously presented configurations, several components are required to make them work. In the simplest case of direct shaft, only bearings, power unit, propeller shaft locking device etc. are needed but in more complex cases there needs to be more components like gears, gear couplings, flexible couplings etc. In ice navigation all of these components need to be able to withstand the experienced ice loads. Typically gears are the most critical components because they transmit high amount of power on a relatively small contact area.

With L and Z configuration bevel gears are used and they can be the most defining components, especially on ice classes. This is because rather small area of the gear tooth conveys the power at any given time and propeller-ice interaction may cause rather high torque excitation which leads to high torque peaks. This means that bevel gears usually limit the possible power output in ice classified units. This is at least the case with the Baltic ice classes but with icebreakers there may be other defining characteristics as well.

Commonly used gear type in azimuthing thrusters is spiral bevel gear. The biggest challenge in the spiral bevel gear design is compensating the deflections as well as possible in all the loading conditions (Häger 2011, p. 4-5). The tooth contact is also much more complicated in general which makes designing the gears more complex. This also means that manufacturing these types of gears demands high precision and skill, but these types of gears also provide some good characteristics that cannot be provided with any other gear types.

Because the spiral bevel gears have their teeth in the spiral shape, they have more teeth in contact and the contact is more gradual between the teeth when compared to their straight teeth counterparts. This means that the spiral bevel gears also have high power transmission capability compared to the size of the gears. For the same reasons spiral bevel gears are much more silent and can be operated under heavy loads and high speeds.

Another vital component that is taken into more careful consideration in this study is flexible coupling. This is because it can reduce the excitation torque and vibration for other components and it is often used to protect the engine or motor. The key things to consider are its damping characteristics and the fact that the flexible coupling changes the natural frequencies of the system.

3.4 Propeller types

Propeller type is also one of the ways to affect the final system characteristics. Most commonly used types in azimuthing thrusters are fixed pitch propeller (FP), controllable pitch propeller (CP) and contra-rotating propeller (CRP). FP or CP propellers are the most common ones, especially in ice navigation. Those propellers can also be pulling type propellers rather that the traditional pushing type.

Typically there are two variations of propellers so that the propeller is either open or ducted. Both of these can be used in same type of conditions but usually their performance differs. Open propeller is not obstructed by the hull or other structures so that the propeller is fully exposed to the surrounding water. Ducted propeller is however equipped with a nozzle that is used to accelerate or decelerate the water flow to the propeller plane. Nozzle influences the thrust performance at different ship speeds.

FP and CP propellers can be equipped with nozzle or they can be open propellers. Both are commonly used and have their own good qualities. Ducted propeller provides more thrust at lower ship speeds but in ice classes it has some bad qualities that sometimes outweigh the benefits. First of all there is a change of blockage when traveling in ice and secondly the open water performance may suffer (Lee 2008, p. 14-18) depending on the vessels intended operations and speed.

FP propeller has a fixed pitch and is best used with variable speed electric motor. Because pitch is fixed the driving engine or motor has to be run in different speeds which can be problematic. Diesel engines have quite limited speed range when they provide maximum torque so situations where the engine needs to be operated outside the nominal speed range are disadvantageous. This is not as much of a problem with electric motors because of the wider rpm range and over torque capabilities. Also FP propeller driven by diesel engine would require much more power if driven in brash ice channels and heavy ice conditions in order to provide the necessary thrust (Lee 2008, p. 9-12).

Because CP propeller has controllable pitch, diesel engines benefit most from these types of propellers. This is because diesel engines can be driven at their nominal output rotational speeds and by controlling the pitch, thrust value can be adjusted. This is also a very good choice for traveling in brash ice channels because of better thrust characteristics (Lee 2008, p. 9, 12-14) and it is also recommended choice with the diesel engine in the rules (Trafi 2011, p10.). However, special care has to be put into the pitch mechanism because of the possible heavy ice loads.

Azimuthing thruster can also have a pulling type propeller which means that the propeller pulls the ship forward rather than the traditional pushing type which pushes the ship forward. Pulling type propeller has more uniform inflow because it doesn’t have the thruster body in the way and consequently has some beneficial hydrodynamic characteristics. But because the thruster body is not protecting the propeller this type of a propeller is also much more exposed to ice contact. These types of propellers are often used for faster ship speeds.

CRP is a dual propeller where two FP open propellers rotating in different directions are installed on coaxial shafts. The second propeller absorbs some of the rotational energy of the first propeller thus providing better efficiency. It is basically a combination pulling type propeller as the first propeller and pushing type as the second propeller. These kinds of propellers are quite special designs and are usually not considered to be used in heavy ice conditions. This type of propeller is also meant for high speed ships and it can be more fuel efficient because some of the rotational energy is absorbed.

In this study pushing type FP and CP propellers are examined later on. There isn’t that much difference how the propeller loads are thought to be occurring in those propeller types and the biggest difference occurs between open and ducted propeller. These and other design variables that affect the propeller ice loads are studied with an example thruster unit.

4. PROPELLER LOADS

Typical propeller loads are either hydrodynamic or mechanical and are caused by contact or non-contact loads. Normally on open water, hydrodynamic case is governing the design but in ice waters the case is different and rather complicated. This is because when considering propeller-ice interactions there are several ways and stages how ice block may cause loads on propeller and these loads are highly stochastic in nature. Furthermore the loads are dependent on the ice properties and manifestation.

Ice can cause both contact and non-contact loads on propeller blades and the loads are also conveyed to other parts of the propulsion machinery. This usually happens during icebreaking as some crushed ice blocks travel submerged in the water along the vessels body. Typically both load conditions are present as the submerged ice block first travels on the upstream of the propeller causing hydrodynamic disturbances and then it might hit the propeller blades causing mechanical contact loads. It may cause a blockage as well on a ducted propeller if the ice block or blocks get stuck on the nozzle. This kind of a scenario can also happen with ice blocks floating in the ice channel but there are also numerous other ways and combinations how the ice interaction occurs.

Non-contact loads also occur when ice block restricts or blocks the water flow to the propeller. These kinds of loads are more common and more detrimental on nozzle propellers but they affect open propellers as well. This is because nozzle blockage can cause high unsteady and prolonged hydrodynamic loads. Non-contact loads don’t usually cause the maximum torque peaks but have to be taken into account in propeller blade design because of the cavitation.

Contact loads are mechanical loads and they are either caused by ice impacts or ice milling. Impact loads can affect any part of the propeller as those can hit the propeller blades, hub or strut for example. Typical case that causes torque peaks on propellers is ice milling according to full-scale measurements (Koskinen et al. 1996, p. 18). Milling can be considered an impact like contact load as a propeller blades cut the ice and ice breaks into small particles (crushed ice). The difference is that milling impacts happen multiple times during a relatively short period of time.

In this study the theoretical basis for ice block milling is determined according to Finnish-Swedish ice class rules. These rules are based on long term measurements and numerical models that were used in creating the formulations. (Browne and Norhamo 2007, p. 9) The equations for ice excitation torque are also similar in the Polar classes.

Finnish-Swedish ice class rules are result of a long term work where many full-scale measurements, ice pressure measurements, simulations and calculations are combined into simplified formulations. (Koskinen et al. 1996 p. 18) The ice class rules provide maximum propeller excitation torque and milling process load cases. The resulting maximum is the maximum encountered ice load caused by either non-contact or contact load. Both of these are included into the equations and are to be applied into time domain simulation. The rules also tell the amount of expected ice load cycles which is directly applicable to the fatigue stress calculations provided that the load spectrum is first generated.

4.1 Excitation torque

In Finnish-Swedish ice class rules (2010) it is considered that ice block of size $H_{ice}\cdot 2H_{ice}\cdot 3H_{ice}$ hits the propeller and causes the maximum encountered excitation during the propellers lifetime. The design ice thickness H_{ice} in each ice class is presented in Table 2 (Trafi 2010, p. 27).

Table 2. *Design ice block thickness in Finnish-Swedish ice class rules.*

Ice class	**Design ice thickness H_{ice} [m]**
ICE-1A super	1.75
ICE-1A	1.5
ICE-1B	1.2
ICE-1C	1.0

Because the ice classes are to be operated in different kind of conditions and varied amount of icebreaker assistance, the design ice block thicknesses also differ. For example a 1A super ice classed vessel is thought to be operated in difficult ice conditions and even in level ice without icebreaker assistance, so the propeller may also encounter bigger ice blocks. (Trafi 2010, p. 4)

As the ice block enters the propeller and propeller-ice interaction occurs, there are torsional ice excitation loads on propeller. These loads differ depending on what kind of propeller design is being used, ducted or open, and other propeller properties. Maximum encountered excitation torques are defined as (Trafi 2010, p. 35-36)

Open propeller:

$$T_{max} = 10.9 \cdot \left[1 - \frac{d}{D}\right] \cdot \left[\frac{p_{0.7}}{D}\right]^{0.16} (nD)^{0.17} D^3, \text{when } D \leq D_{limit} \tag{1}$$

$$T_{max} = 20.7 \cdot \left[1 - \frac{d}{D}\right] \cdot \left[\frac{p_{0.7}}{D}\right]^{0.16} (nD)^{0.17} D^{1.9} H_{ice}^{1.1}, \text{when } D > D_{limit} \tag{2}$$

Ducted propeller:

$$T_{max} = 7.7 \cdot \left[1 - \frac{d}{D}\right] \cdot \left[\frac{p_{0.7}}{D}\right]^{0.16} (nD)^{0.17} D^3, \text{when } D \leq D_{limit} \tag{3}$$

$$T_{max} = 14.6 \cdot \left[1 - \frac{d}{D}\right] \cdot \left[\frac{p_{0.7}}{D}\right]^{0.16} (nD)^{0.17} D^{1.9} H_{ice}^{1.1}, \text{when } D > D_{limit} \tag{4}$$

where d is propeller hub outer diameter, D is propeller diameter, $p_{0.7}$ is propeller pitch at $0.7R$ radius and n is propeller rotational speed in rps. D_{limit} is $1.8 \cdot H_{ice}$ and T_{max} has unit kNm. This formulation is a result of joint Finnish-Canadian research project JRPA#6 and it was derived from the numerical model which was used to study blade design loads (Browne and Norhamo 2007, p. 9).

4.2 Milling process

The maximum ice excitation is thought to happen with ice block milling. This milling process can happen in reality a multitude of ways because of the block orientation and rotational angle of approach. In the rules there are three described cases how the propeller-ice interaction occurs. In all of the cases, excitation torque T_{max} is used. In the rules the resulting torque for single blade equation is (Trafi 2010, p. 36-37)

$$\begin{aligned} T(\varphi) &= C_q T_{max} \cdot \sin\left(\varphi\left(\frac{180}{\alpha_i}\right)\right), \text{when } 0 \leq \varphi \leq \alpha_i \\ T(\varphi) &= 0, \text{when } \alpha_i < \varphi \leq 360 \end{aligned} \tag{5}$$

where α_i is duration of propeller blade-ice interaction expressed in degrees and C_q is torque coefficient. These coefficients for each case are presented in Table 3.

Table 3. *Ice block milling sequence parameters.*

Load case	**Propeller-ice interaction**	**C_q**	**α_i [deg]**
Case 1	Single ice block	0.75	90
Case 2	Single ice block	1.0	135
Case 3	Two ice blocks (phase shift 360/2/Z deg.)	0.5	45

Total ice torque is a sum of all propeller blades but the phase shift between the blades has to be taken into account. Ice class rules also tell to have 270 degree linear ramps in the beginning and the end of milling sequence. The total number of impacts during ice block milling sequence is

$$N_{mill} = 2ZH_{ice} \tag{6}$$

where Z is number of propeller blades and H_{ice} is thickness of design ice block. The value N_{mill} is rounded towards next higher integer.

The propeller ice excitation is described by propeller blade impacts of half sine shape and it is shown in Figure 1 for 4-bladed propeller and ice classes 1C (left hand side) and 1A super (right hand side). From the shape of the transient load we can also see the used linear ramps in the beginning and at the end. Because the total number of revolutions during ice block milling is a function of design ice thickness H_{ice}, the duration of ice milling sequence is different for each ice class. The number of propeller blades would change the amount of the impacts during milling sequence.

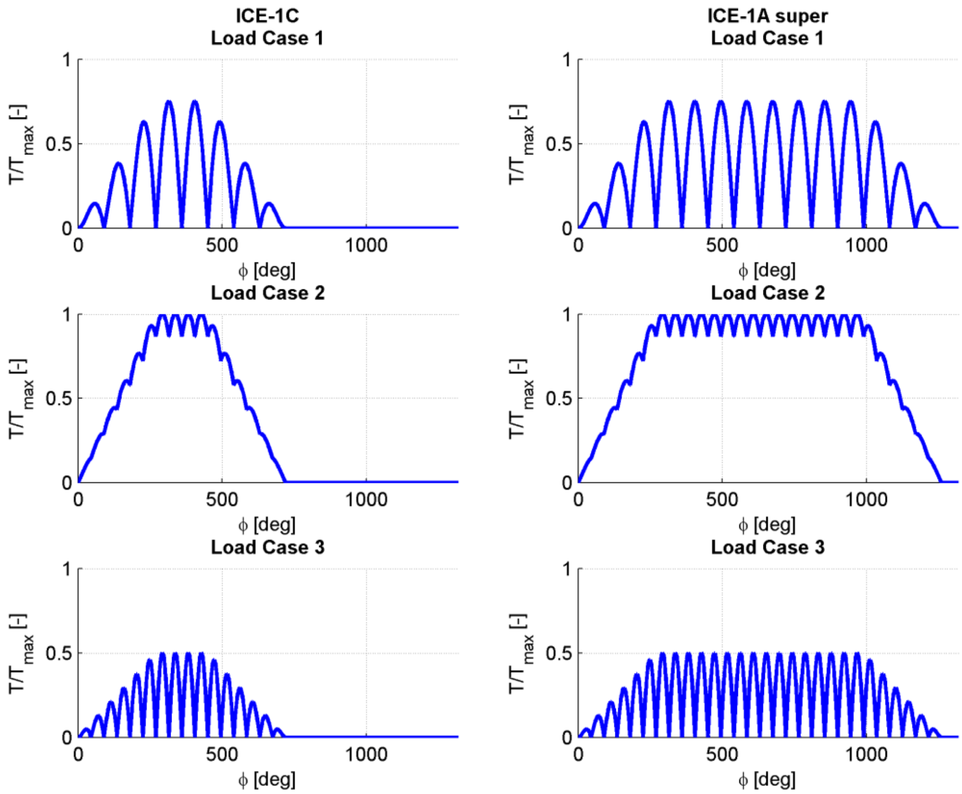


Figure 1. *Ice milling for 4-bladed propeller in ice classes ICE-1C and ICE-1A super.*

Figure 1 shows the difference in ice classes by showing that the amount of ice impacts and the duration of the load increases with higher ice classes. We can calculate that for the ice class 1C, number of ice impacts is 8 and based on Figure 1 the duration of these is 720°, two revolutions. Similarly for ice class 1A super the number of impacts is 14 and the duration 1260°, 3.5 revolutions. These make sense because those are consecutive blade hits and because 4-bladed propeller was used, 8 hits mean two revolutions. In case of ICE-1C each propeller blade hits the ice block twice and in the case of ICE-1A super propeller blades 1 and 2 hit the ice block 4 times and blades 3 and 4 hit the block 3 times.

Because the ice milling is described by a variable describing rotational angle, it means that the actual time of interaction depends on the propeller rotational speed. Consequently as the ice milling happens over time, it is easiest to get results via simulation. This means that the milling process should be implemented in a simulation model for each load case. It is also needed in order to evaluate the transient ice excitation effect on the propulsion machinery components.

These ice loads are applied on propeller at the maximum continuous rating (MCR) and can affect the propulsion system in several ways. Ice milling causes extra torque peaks and as the main engine tries to maintain the propeller speed, the engine rpm drops. In the worst possible situation engine might stall. Ice loads also cause vibrations which may excite system's nominal frequencies. (Lecourt and Zahn 1985, p. 182) Because there are three different evaluated cases there is also higher chances that some load case excites these natural frequencies.

Although the rules basically describe only occasional ice block milling, there exist double-acting ships that use bow thrusters and are subject to more frequent propeller-ice loading conditions. This has to be taken into account in the propeller by designing it according to all the class requirements. This at the minimum means that the amount of ice load cycles propellers encounter is higher in this kind of practice.

4.3 Ice load cycles

Because the ice induced loads are highly stochastic in nature, there is some distribution between the highest expected torque load and the nominal condition. The total amount of these loads is generally taken into account by calculating the total amount of load cycles occurring during the lifetime. This is then applied to generate a load spectrum for the fatigue analysis, two-parameter Weibull distribution is typically used. Equation for the total amount of ice loads N_{ice} is (Trafi 2010, p. 33)

$$N_{ice} = k_1 k_2 k_3 k_4 N_{class} Z \tag{7}$$

where Z is the amount of propeller blades, N_{class} is the reference number of loads for the ice class and factors k are related to propulsion and chosen propeller. Factors k are presented in Table 4 and the reference number of loads is presented in Table 5.

Table 4. *Factors for calculating number of load cycles.*

Propeller location factor k_1	Centre propeller $k_1 = 1$	Wing propeller $k_1 = 1.35$
Propeller type factor k_2	Open $k_2 = 1$	Ducted $k_2 = 1.1$
Propulsion type factor k_3	Fixed $k_3 = 1$	Azimuthing $k_3 = 1.2$
Submersion factor k_4	$k_4 = \begin{cases} 0.8 - f \; when \; f < 0 \\ 0.8 - 0.4 \cdot f \; when \; 0 \leq f \leq 1 \\ 0.6 - 0.2 \cdot f \; when \; 1 < f \leq 2.5 \\ 0.1 \; when \; f > 2.5 \end{cases}$	where $f = \dfrac{h_0 - H_{ice}}{\frac{D}{2}} - 1$

The variables in Table 4: H_{ice} is the design ice block thickness, h_0 is the depth of the propeller centerline at lower ice water line and D is the diameter of the propeller.

Table 5. *Reference number of ice loads in each ice class.*

Ice class	**ICE-1C**	**ICE-1B**	**ICE-1A**	**ICE-1A super**
N_{class}	$2.1 \cdot 10^6$	$3.4 \cdot 10^6$	$6 \cdot 10^6$	$9 \cdot 10^6$

All of the rules and values presented in this chapter are based on Trafi (2010) Finnish-Swedish ice class rules 2010 p. 33-37. These are applied to the calculation process that is done to evaluate loads on different components with simulation. The simulation is based on torsional vibration analysis (TVA) principles and it is done in time domain. The amount of load cycles is used in gear calculation for the distribution of ice loads for the lifetime of the gear.

4.4 Torsional vibration analysis

Torsional vibration occurs when torsional moment is applied to the system and the shafts start twisting as they rotate. This happens especially with reciprogating machinery like diesel engines because they generate fluctuating torques which may excite system's natural frequencies. In propulsion machinery propeller can also cause fluctuating torques as the propeller is rotating in the water. It generates excitation torques because of the non-uniform flow on the propeller plane and hydrodynamic pressure changes in the water. Other components in the driveline can also generate fluctuating torques because of irregularities in materials and contacts, slipping and transmission errors for example.

Depending on how the system is operated these kind of excitations can be quite harmful. This is because the propulsion machineries are driven usually in a range of different rotational speeds and some nominal frequency can be excited. This is harmful because then the dynamic torque can be amplified and it may cause fatigue damage and wear in rotating components. (Feese and Hill 2009 p. 213-215)

Those kinds of excitations are usually continuous and periodic but the ice load excitation cases that were presented earlier are transient, short term, impulse like excitations. Other transient loads occur for example during start-ups and clutching of heavy components. This means that there are some key differences in how ice excitations are taken into account. The basic principle is still the same because the whole analysis is based on the equation of motion that can be written as

$$\boldsymbol{J}\ddot{\varphi}(t) + \boldsymbol{C}\dot{\varphi}(t) + \boldsymbol{K}\varphi(t) = T(\mathrm{t}) \tag{8}$$

where the $\boldsymbol{J}$ is inertia matrix, $\boldsymbol{C}$ is damping matrix, $\boldsymbol{K}$ is torsional stiffness matrix, φ is the displacement vector and T is the vibration excitation torque vector. The equation (8) can be written into n amount of equations which correspond to the number of nodes or lumped masses in the system. (Magazinović 2014, p. 506)

If the vector T is periodic, these types of vibrations are so called steady-state vibrations and those can be solved in frequency domain. It is used to solve resonance frequencies, vibration amplitudes and angular phases which are often presented graphically. (Magazinović 2014, p. 506-507) However, frequency domain analysis is only valid for the systems where the acting excitations are independent on time and the system responses remain periodic or harmonic.

In this case when the ice loads are transient, dependent on time, simulation needs to be done in the time domain. This requires the system model to be solved with numerical time integration. This generally means that the differential equations of the model are solved based on the simulation time and given time step. There are multiple different methods how the time integration can be done and those are utilized in commercial softwares.

In this study Matlab Simulink variable time step ode 23t solver is used. The chosen time step solver uses trapezoidal rule with free interpolant. (Matlab Help 2015) It's a linear numerical integration method for solving differential equations and based on how well the result of the next iteration step fits the error tolerances, the time step is varied. This means that the time step is shorter for rapidly changing results because the integration method can't approximate the next iteration step result accurately enough otherwise. The accuracy also depends on the loads and the complexity of the simulation model.

Because these ice loads do not only affect the propeller but the whole propulsion machinery, the system needs to be presented in a simple enough way so that the calculations can be done in a reasonable amount of time. Normal engineering practice in this kind of case where there is a complex system is to simplify it into lumped mass-elastic system. This simplified mass-elastic model is used as a simulation model and harmful running speeds or peak loads are solved through simulation.

This of course means in this case that the whole thruster needs to be basically designed before any calculations can be done so that every part can be reduced into the mass-elastic model. Usually the mass elastic data can only be verified at the later stages of the design which is problematic especially in the sales offer calculations. Even in sales offers the more accurate model would give more accurate results which would possibly results in more competitive characteristics in sales. But if we have some kind of a reference point to start from, it reduces the amount of needed iteration in the design and increases the accuracy of the preliminary calculations.

In this study we're interested in the highest peak loads because all of the propulsion machinery components need to be designed so that they're able to withstand those. This is done by implementing the mass elastic model, equation of motion (8), ice excitation and other nonlinear behavior of the components into a simulation model. This model is then solved at each time step and the peaks loads are evaluated.

Common practice is that the peak torque value is converted into an application factor which means that the maximum torque is divided by the nominal torque. The application factor can then be used for example in the static design of a component. This makes it also easier to understand how much higher loads the components need to withstand compared to the nominal condition.

5. CALCULATION PROCESS

The main idea in the process is that each propeller unit component has to be able to withstand the encountered loads during its lifetime. This is done by first calculating maximum excitation torque (equations 1-4) and then doing a torsional vibration simulation based on a design project. The different milling load cases (equation 5) are applied into the simulation and the maximum loads for the different components are evaluated. Then the critical components are designed accordingly with the chosen classification society rules.

The most restricting factor in the components is usually the fatigue strength. In open water cases the components are usually designed for infinite life but in ice classified thrusters there is additionally an ice load distribution. This distribution takes into account the encountered higher loads that may cause fatigue damage. The amount of loads in this distribution is calculated with equation (7) and is applied in forming the ice load spectrum.

In the preliminary calculations for example in sales offers, there are a lot of different unknown variables that won't be confirmed until much later stage of the design. Therefore usually some old project is chosen as a reference point. This of course is quite inaccurate as there might be significant differences in the design afterwards that may cause problems with the promised sales values. This is one of the biggest problems in the current rules from a propulsion unit supplier standpoint.

Normally ship manufacturer requires certain power or thrust for a new ship and the thruster supplier would like to make an offer from their current, well-established unit selection. This is because it's better to have limited set of parts and units design-wise and manufacturing-wise. In addition classification process for new parts can be time consuming and it might be costly because of all the tests and calculations. Also, parts have to be classified for all the needed societies independently so it would be beneficial to use same gears and other parts that fulfill the requirements for all the societies in both open water and ice classes. This is also easier while considering the ice load calculation because fewer variables need to be estimated and the existing component designs can be utilized.

Although there are many parts that are subject to ice loads, gears are often a limiting factor in azimuthing thruster designs (especially in ice classes), so a good and easy-to-use simulation and gear calculation process would be optimal. This is hard to do because as a unit is being initially offered for a customer, not much can be said about the thruster design yet and therefore there are several uncertain parameters in the ice calculations itself. Also sales team, which needs this kind of tool the most, may not have all of the technical knowledge needed for such calculations. This means that a lot of different factors should be taken into account but at the same time the calculation process should be simple enough for everyone to understand and use.

In this study the power is the main question because if the customer has certain thrust demand, it can be derived into a corresponding power value. Then a suitable propeller unit is chosen and the power is derived to the needed nominal torque. Maximum ice excitation torque is additionally affecting the system but because its magnitude is only related to the propeller speed, it stays nearly constant in the simulation results. Therefore total maximum torque in ice can be affected by lowering the nominal power. This lowers the total maximum experienced torque. A balance needs to be found so that the gears are valid for classification of the chosen society. In this study the chosen society is DNV and the results are also compared to DIN 3991 (1998) gear calculation standard, which is used by other classification societies.

If the final value for the power is not enough then bigger thruster unit needs to be chosen. The most optimum situation would be that the propeller unit would meet the criteria just barely as it would be financially the best possible offer the thruster supplier can make. If however the power is just barely too low, bigger unit has to be offered and it may cost more, so it wouldn't be as lucrative for the customer.

One unit is taken to a closer inspection from Rolls-Royce selection and a calculation tool is developed for it. Many other azimuthing thruster units have similar power transmission structure so the developed process can be applied to other units as well but if the structure differs much, some changes need to be done in the simulation model and to the Matlab script itself.

5.1 Current process

The current process starts by assessing ice excitation in MCR condition. This means calculating maximum excitation torque according to equations (1)-(4). Ice milling is then simulated with the simulation model and as a result, one gets application factors for gears and other components. These are then used in gear calculation and evaluation if the gears are valid to use or not. If the gears have too low safety factors then the nominal power needs to be lowered and the process needs to be done again until safety factors are on an acceptable level.

Current calculation process is presented in Figure 2. There are several problems related to this process because some iteration may be needed and the iteration is done manually. Another thing is that the classification societies have different rules that they use which makes the process harder to adopt. The ice excitation torque is calculated the same way for Baltic ice classes in almost all of the approved societies but gears are evaluated based on different standards and rules. DNV, which is studied more here later, has their own rules and some others use DIN standards for example. This means that there are usually a couple of different calculation programs or templates that are used which makes the process even more complex.

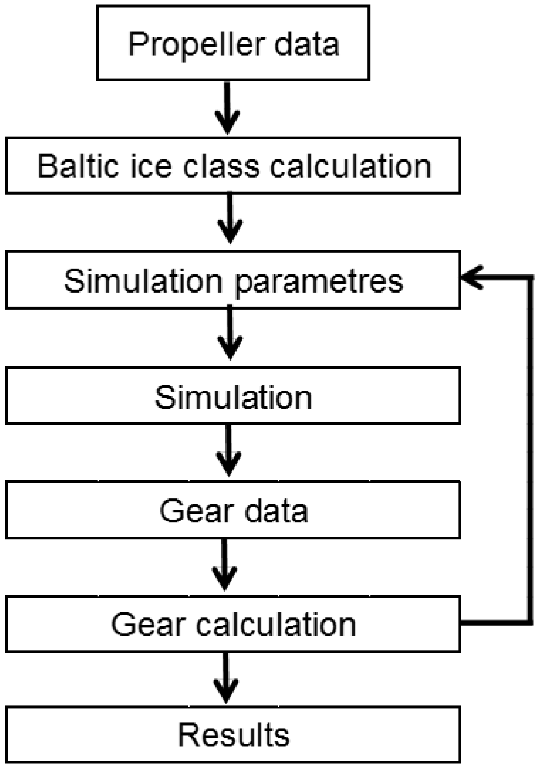


Figure 2. *Ice class calculation process.*

In this kind of process user inputs are needed in propeller data, simulation parameters and gear data. In addition iteration needs to be done manually if the simulation and gear calculation is done in different programs or templates. This can also cause accidental mistakes if some of the iteration parameters are not updated during the iteration.

Usually the iteration process means that the unit’s power needs to be reduced which means lower nominal torque. This is done because the ice excitation torque is not affected by the unit’s nominal torque but is basically in addition to this. Lowering unit’s nominal power, which corresponds to nominal torque, reduces the total maximum load. This leads to lower loads and higher safety factors for the gears. This is done currently manually and takes a lot of effort and time. It also requires much more knowledge from different gear calculation rules and standards.

Because all of the above is based on well defined rules, it could be simplified and put in a single program with the simulation model. This would make it possible to do calculation and iteration automatically which in turn would ease the calculation process for many people. This of course needs to be done so that the program would be relatively easy to update and so that all the possible choices for ice classes and gear rules are available.

A well working simulation model is a key to a good procedure and with Matlab Simulink it is relatively easy to do for a lumped mass model. However it still needs to be thoroughly thought out in order to make it adaptable for as many cases and units as possible. Simplification needs to be thought out carefully so that the results are applicable for their purposes. Some parts of the simulation also need to happen automatically and all the needed results should be easily available.

5.2 Simulation model

The azimuthing thruster unit that has been taken to closer inspection is Rolls-Royce US 305 and its simulation datasheets can be seen in appendix 1 and 2. These have been used as a basis for the created simulation models. The propeller unit has Z-drive and can be driven by electric motor or diesel engine, and it can be equipped with FP or CP propeller. The propeller can furthermore be either open or ducted.

There are slight differences in the diesel and electric driven machineries. Usually diesel engine propulsion unit requires some kind of intermediate shafting between the upper gear and the flexible coupling. It’s because it may need some space around it for maintenance operations or because of lack of space so that the engine needs to be placed further away. This shafting also varies from project to project. Electric motor can usually be placed right next to the propulsion unit and additional shafting is unnecessary.

The unit’s power drive is reduced to lumped mass-elastic system to make the calculation faster and easier to model. Simulation model is done in Matlab Simulink and the lumped mass model is represented by inertia, stiffness and damping blocks. These blocks represent the left side of the equation (8) of motion. All of the used blocks in simulation model are basic Simulink blocks or mechanical blocks from Simscape toolbox. Simulation model with electric motor is presented in Figure 3 and it is based on the mass elastic model in appendix 1.

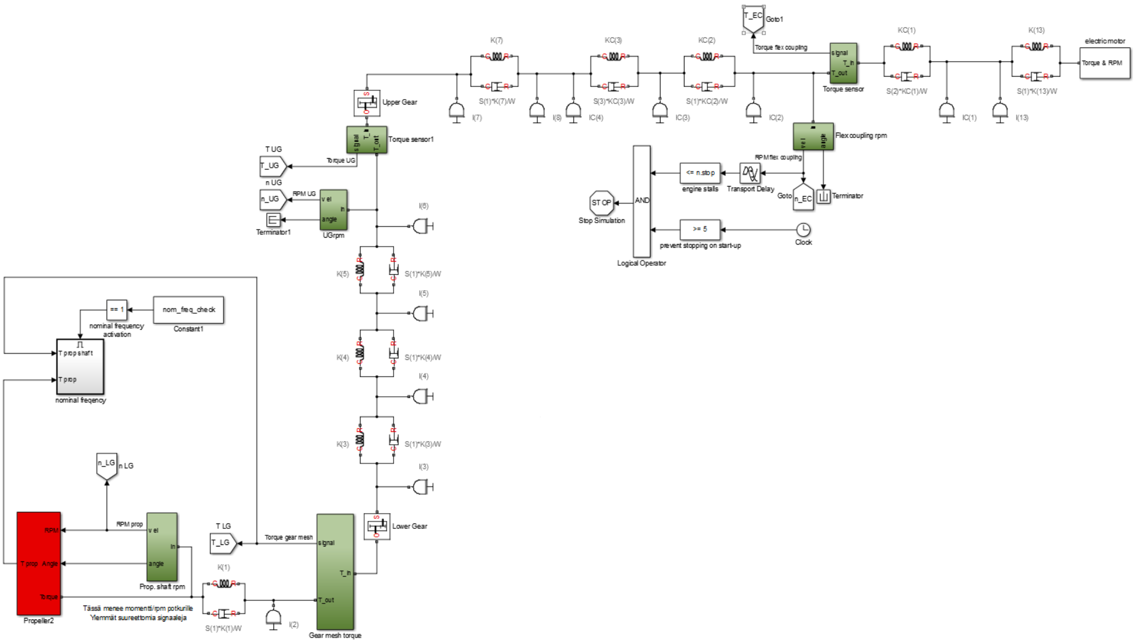


Figure 3. *Simulation model for US 305 with electric motor.*

The diesel model is done in a similar fashion but the engine block is different. The lumped mass model differs a little as well because there is intermediate shafting between upper pinion and diesel engine. This means a couple of more lumped masses on the input shaft before the upper gear. The mass-elastic model for diesel driven US 305 is shown in appendix 2.

Simulation is done in time domain with variable time step solver ode23t and with a maximum time step of 1 ms and relative tolerance of 10^{-4}. The variable step solver changes the time step continuously to increase accuracy when the results are changing rapidly and speeding up the process when results stay relatively the same. (Matlab Help 2015).

Solver ode23t is meant for moderately stiff simulation models and uses the trapezoidal rule with free interpolant in solving the model each time step. The trapezoidal rule uses straight line approximation for the time step and evaluates the value and the error tolerance for it. If the value doesn't fit the error tolerance time step is reduced and the calculation is done again. This error tolerance can be made more accurate by lowering the relative tolerance. (Matlab Help 2015)

The maximum value of 1 ms should be okay for the range of Rolls-Royce selection since the fastest input speed is about 1800 rpm. This means that the shortest propeller-ice interaction time according to equation 6 is 1/8 of a revolution (case 3) resulting in time of 4.2 ms. This load is experienced by the flexible coupling and because the rpm drops during the ice interaction, the duration of the loading condition gets distorted and stretched on a longer period of time. This is also the condition when the variable time step starts to work as the results start to change rapidly and the time step is shortened.

In this case we would like to use this model for preliminary analysis so faster solving is justifiable. Therefore we don't want to increase the level of accuracy or shorten the time step too much. Shorter time steps and decreased error tolerance may give marginally more accurate results but because the model itself is already reduced to a lumped mass model from the continuous masses. Also simulation with shorter time steps takes much longer to evaluate in real time.

This can be seen with the ICE-1A super 60 second simulation with electric motor and all load cases as the simulation would be solved in 60 001 times with fixed 1 ms time steps. The simulation starts from the moment 0 s and the model is solved then for the first time and it is solved for the last time at 60 s mark. The variable time step solver with 10^{-3} relative tolerance solves the model 61 767 times so the time step is shortened at some points in order to maintain accuracy. These additional calculation steps seem to be during the ice milling when the results change rapidly. If the relative tolerance is lowered to 10^{-4} the model is solved at 68 423 times but the accuracy doesn't seems to change much. With 10^{-3} tolerance the elastic coupling maximum torque is 51.525 kNm and with 10^{-4} tolerance 51.4106 kNm. The level of tolerance seems to affect 0.22 %.

If the maximum time step is changed to 0.1 ms with 10^{-3} relative tolerance the model is solved 617 480 times and the result for the elastic coupling is 51.7702 kNm. This means 0.48 % difference with the settings used in this study but the simulation takes much longer in real time. Shorter time steps are justifiable when propeller or engine excitation is examined but for these sales offer purposes, the 1 ms time steps are accurate enough.

In the model, propeller block contains propeller torque law, ice load milling cases, equation (5), and propeller excitation according to DNV rules Part 4 Ch. 3 (not part of this study). Milling cases and propeller excitation are made for 4- and 5-bladed propellers and work automatically with the input number of blades. These excitation and milling cases represent the right side of the equation (8) of motion, transient excitation torque vector. Propeller torque law defines the propeller torque need at certain rotational speed and additionally it represents propeller damping (DNV 2011, p. 43).

An additional block inside the propeller was added to control when ice loads occur. The ice loads are controlled to be either on or off and the ice load starting point is associated automatically with MCR condition but there is also an option to set the ice load starting points manually if necessary. Simulation time is now a key factor since if it is too low, the load cases may not occur in time and the load case doesn't show in the results. Setting simulation time is however relatively simple and it doesn't require as much understanding as loads occur automatically when certain operational limits are met. This addition also enables to run all the ice milling cases in the simulation run.

Damping is modeled as relative damping related to lowest excitation frequency of propeller shaft. This frequency value can be evaluated with propeller excitation or it can be solved by some other means. Absolute damping is not used because it is normally used with systems that have constant speed and in this case, it cannot clearly distinguish vibration and change in speed (DNV 2011, p. 41).

Electric motor block contains rotor inertia but in more detailed models rotor stiffness is usually modeled as well. In most preliminary cases, available torsional vibration calculation data is limited to only the total mass moment of inertia and therefore is simply modeled with one inertia block. This approach is also better considering that total mass moment of inertia is usually listed on motor catalogues and it is therefore better available. The motor model provides filtered torque based on the propeller requirement with small lag and it is capable of providing over torque at a set rate and level.

Diesel engine is also modeled with one total mass moment of inertia and by using rotation speed–torque curve. This data is also much more easily available because the more detailed torsional vibration analysis (TVA) data for the engines is usually proprietary and the manufacturers don't want to give it out. The model also has a filtering system which somewhat takes into account the diesel engines firing delays (DNV 2011, p. 44).

The simplification of the engine to only one total mass of inertia also shouldn't have huge impact on the results (DNV 2011, p.43) and this is also shown in Figure 4. The simplified total inertia results are compared to a data normally used in engine torsional vibration analysis. The TVA data contains inertias and stiffness's of individual components of the engine, like cylinders and crankshafts. All of the other system parameters are kept constant.

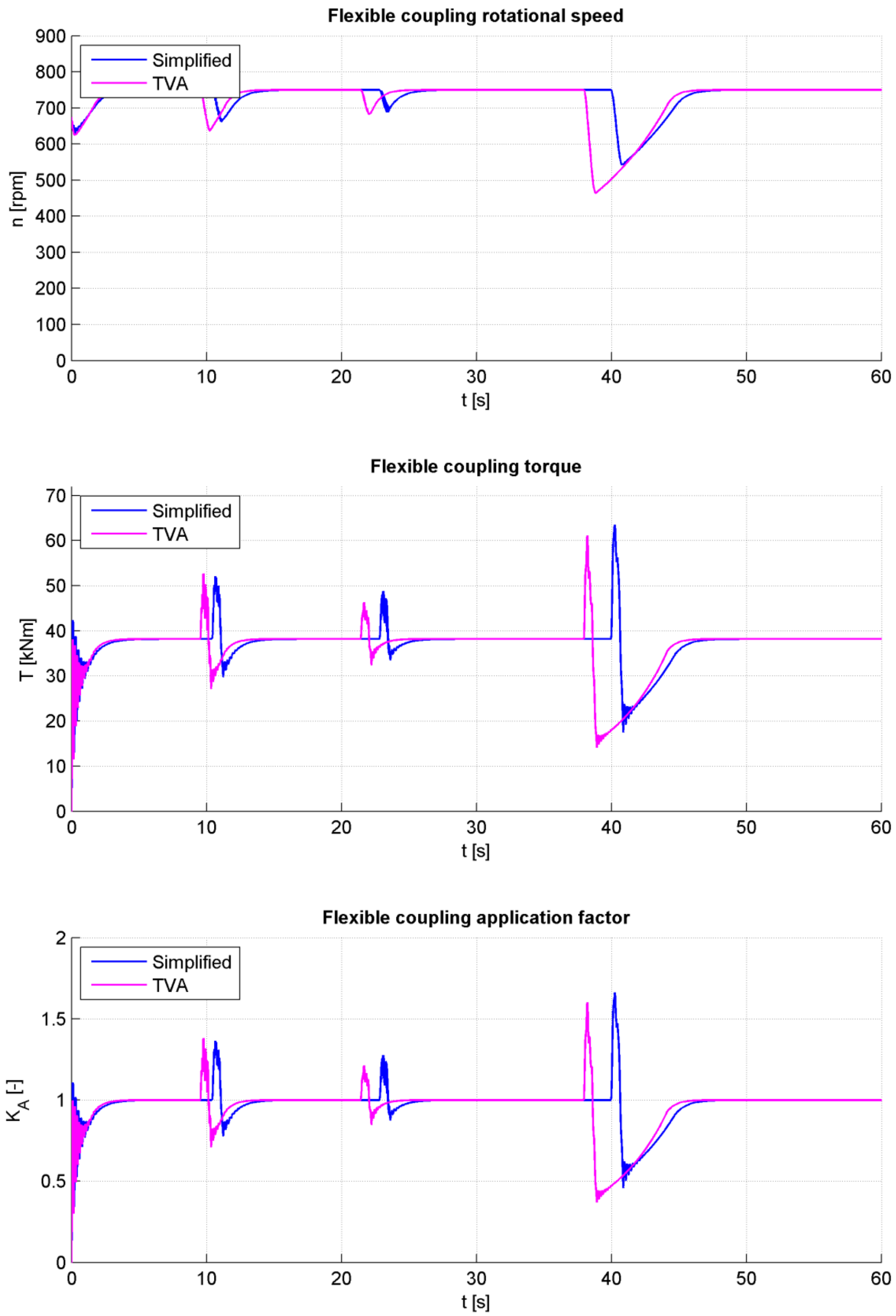


Figure 4. *Comparison of results from engine TVA data and simplified total mass moment of inertia data.*

The TVA data is taken from Torsional vibration analysis MAK 9 M25C (2011) and the total mass of inertia is calculated from that with reduced inertia method (Nestorides

1958, p. 206-207) so that the results are comparable. Figure for flexible coupling is presented because it is the closest component to the engine in the system.

The calculated total mass moment of inertia (558.5 kgm^2) is higher than the minimum presented in M25 C propulsion project guide (2012) (435 kgm^2). This tells that expertise is needed in structural dynamics in order to do this kind of calculation and this also shows that the online catalogues have design values only for preliminary calculations. The final values for each project need to be confirmed by the engine manufacturer because the engines are usually tailored for suitable for each purpose.

The torques for simplified and TVA data model for upper gear are 67.373 kNm and 65.344 kNm respectively. Similarly for flexible coupling the results are 63.477 kNm and 61.162 kNm and for the lower gear 239.816 kNm and 236.231 kNm. This shows that the results are quite close but the simplified model seems to give higher torque and smaller speed drop for all of the inspected components. This is because engines have little bit of damping in them which cannot be taken into account with only one lumped mass but the results are really close.

This should be quite okay at least for preliminary calculations considering that there are many other uncertainties as well and because the total mass moment of inertia seems to give higher loads than the TVA data. The higher torque results give a little bit more safety for the preliminary calculations but if more detailed data is available, it might be better to use that. It would seem better at least in the design phase so that the harmful resonance frequencies can be found out and better understanding of the final system dynamics can be achieved. These factors are at least important when choosing a flexible coupling.

In this study there is no general interest on what happens when the coupling is engaged or disengaged. This could be done by modeling the engagement with additional blocks that control the torque transmission through the component (Pollari 2012, p. 19-20) but it is not done in this model. Flexible coupling is simply represented by the inertia, stiffness and damping values.

Flexible coupling properties are generally highly dependent on the loading conditions like torque and operating temperature. This affects flexible coupling’s stiffness and damping properties and makes it harder to model as these properties are highly nonlinear. Because of these nonlinear properties, it is really hard to describe its operation accurately and it is described with static values only as is recommended (DNV 2011, p. 41). Usually the effect of cold and warm coupling is studied independently by reducing the damping and stiffness values from nominal cold values (Vulkan Technical Data 2014, p. 24).

Torque and rotational speed sensors are Simscape blocks for ideal sensors and they are used to export the simulation data on upper and lower gears and flexible coupling. Maximum value is then taken for each component and application factors are calculated.

The developed simulation model is used for the purposes of this study and it is used in determining maximum gear and coupling loads. All of the required results are exported to Matlab workspace and plotting, data handling and further calculations can be carried from there on.

5.3 Gear calculation

As we get the component load results from the simulation we can start designing and assessing the individual components. The key components are spiral bevel gears that are used with thrusters as those are the most restricting components for total torque.

In this case where there are several classification societies that are responsible for classification of ships and propulsion machineries, the requirements between these societies also differ a little. This can be seen as different standards and rules used for the component design. DIN 3991 (1988) standard is commonly used but for example DNV have their own gear rating rules.

There are differences between the two approaches but because both of these rules try preventing breakages and damages on the same mechanical components, the differences are mostly in the way equations and different factors are presented. Both of these are based on the same principle where bevel gear midsection is converted into equivalent helical cylindrical gears.

These gear calculations need to be easily applied in the new calculation process so that every possible combination of gears can be chosen and their pre-existing gear data is applied to the calculations automatically. The following gear calculation procedures were implemented as Matlab functions where the gear data, simulation results and load cycle calculations are used as inputs.

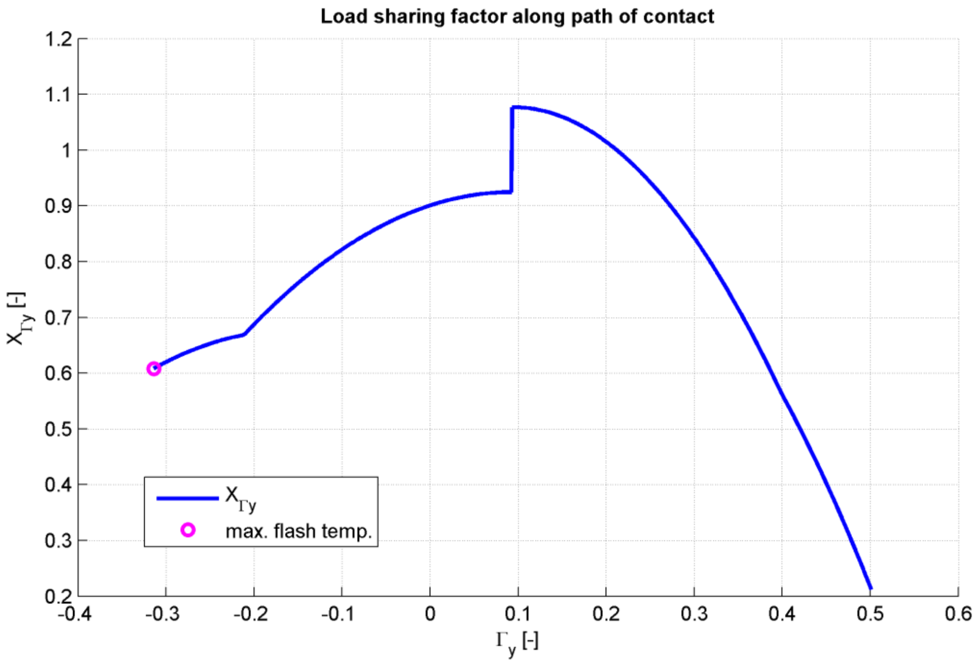
5.3.1 DNV marine gear rating

Gear calculation was done according to DNV Classification Notes 41.2 (2012) and it was done for the Klingelnberg cyclo-palloid bevel gear calculation purposes only. Those rules however define also other gear types. The rules take into account contact stresses, subsurface fatigue, tooth root stresses and scuffing. Safety factors for all of these cases are evaluated independently. Ice load spectrum is added on top of this according to DNV Classification Notes 51.1 (2011) which means that the safety factors are calculated by taking into account the whole ice load spectrum.

Contact stresses take into account surface durability and subsurface fatigue. Surface durability damage can appear as pitting, spalling or case crushing for example. Subsurface fatigue (also known as tooth internal fatigue fracture) calculation takes into account the breakages initiated below the surface on the surface hardened gears. (DNV 2012, p. 5, 23) Both the contact stress and the subsurface fatigue have their own safety factors.

Tooth root stresses are calculated against tooth root cracking and yielding. These are calculated based on gear rim thickness and a high safety factor is used to ensure safety against breakage. (DNV 2012, p. 30-31) Usually damage on the tooth root means the end of the gear's life.

Scuffing is also considered and it means that the teeth are cold welded together because of a high loads and sliding velocities that break the oil film. This welding happens at certain flash temperature caused by the high loading condition. The flash temperature is calculated from the starting point of pinion root and wheel tip to the end of pinion tip and wheel root contact point. Maximum temperature is solved from this contact path and it is called flash temperature. Flash temperature and the base temperature of the condition when the high load occurs are used to determine the safety factor against scuffing. An example of the contact path and flash temperature calculation result can be seen in Figure 5.



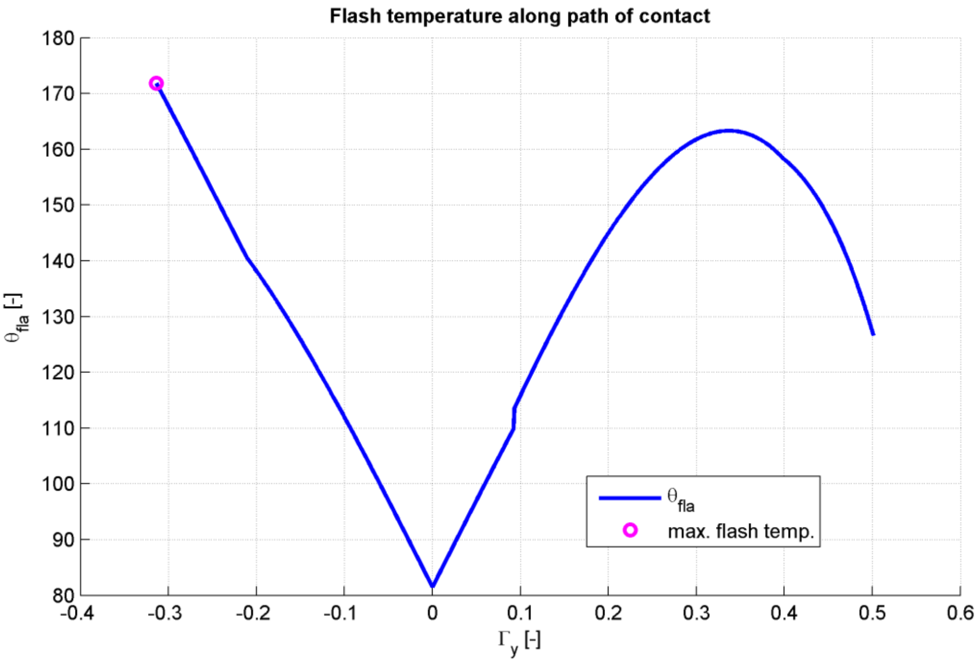


Figure 5. *Example of scuffing load results for load sharing factor and flash temperature.*

Because the gear calculation is based on spiral bevel gear midsection some more exact things are not considered like that the maximum load may not be exactly in the middle of the tooth at all loading conditions. The load area however can be validated for example with full-torque tests and then the calculation with this way is valid. Another way is to use Becal which is approved by the DNV classification notes 41.2. Becal is a bevel gear calculation tool that is used in analyzing tooth contact loads and it may be handy when the tooth contact patterns need to be further optimized.

The ice class calculation is based on distributing ice load amplitudes on the expected ice load cycles and calculating safety factors. This is done according to DNV Classification Notes 51.1 (2011). Weibull distribution is used and it is divided into discrete equally spaced columns on a logarithmic scale. The columns represent the load spectrum conservatively so that the loads and the amount of load cycles are approximated higher than directly with the continuous two-parameter Weibull distribution. This way of describing the load spectrum is also more conservative if fewer bins are used. This is explained better later on in chapter 6.1.

This load spectrum is then applied to surface durability, subsurface fatigue and tooth root stresses where safety factors are calculated for all of the load bins independently. For contact stresses and tooth root stresses Miner damage sum is also calculated from the load spectrum in order to prevent cumulative fatigue damage. Example of these spectrums is given in Figure 6 and Figure 7 respectively. The figures have the calculated stresses for both pinion and wheel which don’t differ that much in this case because those are made from the same material. Discrete distribution steps can also be seen quite clearly on these figures.

The used values for calculations are on the top-right corner points of the distribution so that the higher amount of load cycles is used with a higher load than a continuous spectrum would give. But as we can see from the figures, the most important point is near the endurance limit on the S-N curve, near the 10^8 load cycles. This is the case for both pinion and the wheel and it is a defining point if there’s a high amount of load cycles but in some cases Miner damage sum that is calculated for the whole spectrum can also be restrictive. Scuffing is calculated with maximum ice load so the ice load spectrum doesn’t affect it that much.

However contact stress or tooth root stress is not the deciding factor in the ice classified gears. Subsurface fatigue or scuffing safety factors are the most determining in the ice class calculations and usually one of these governs the allowed power for the propeller unit in ice class. They also have their own safety factors in ice conditions and the validation needs to be done against those.

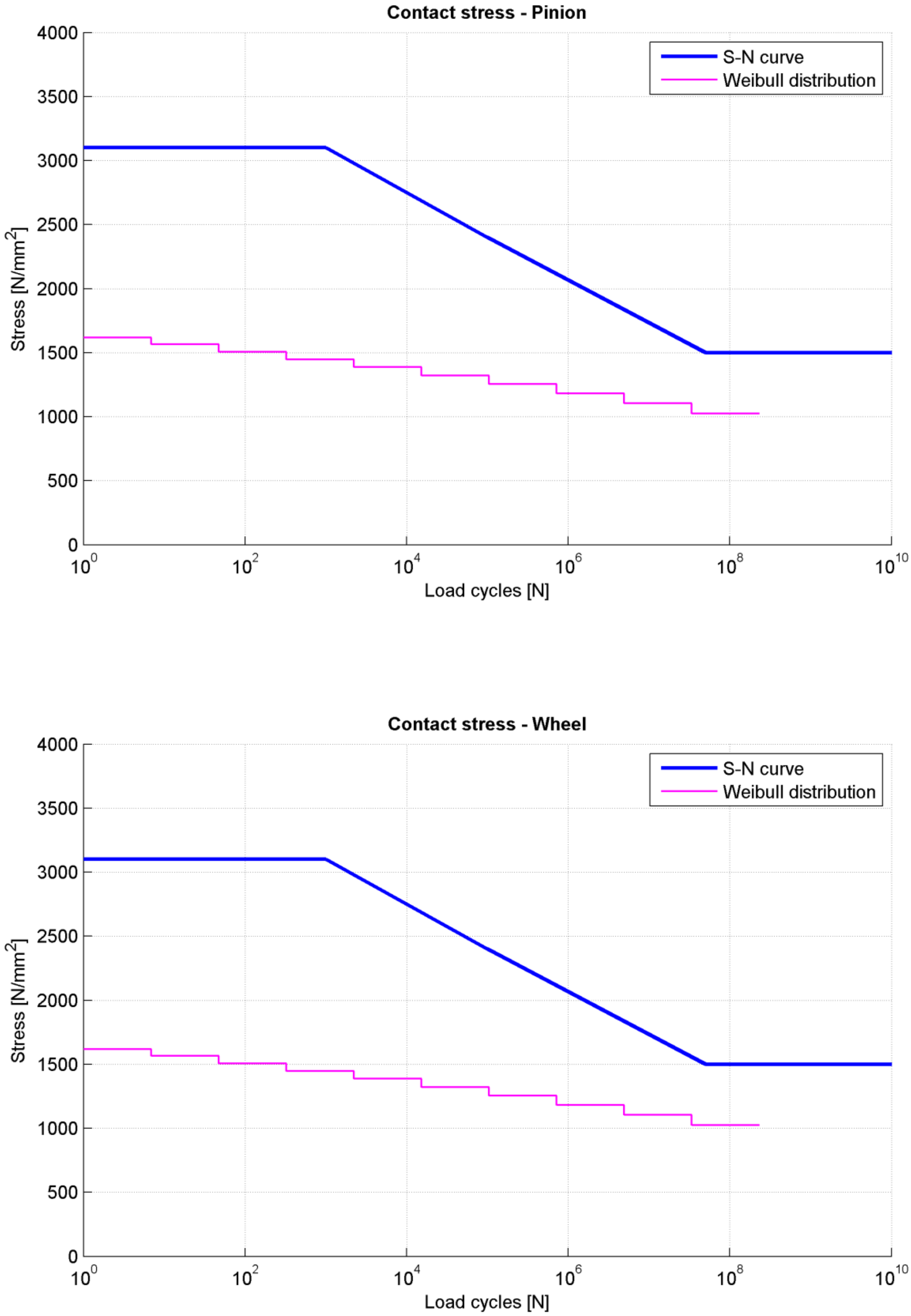
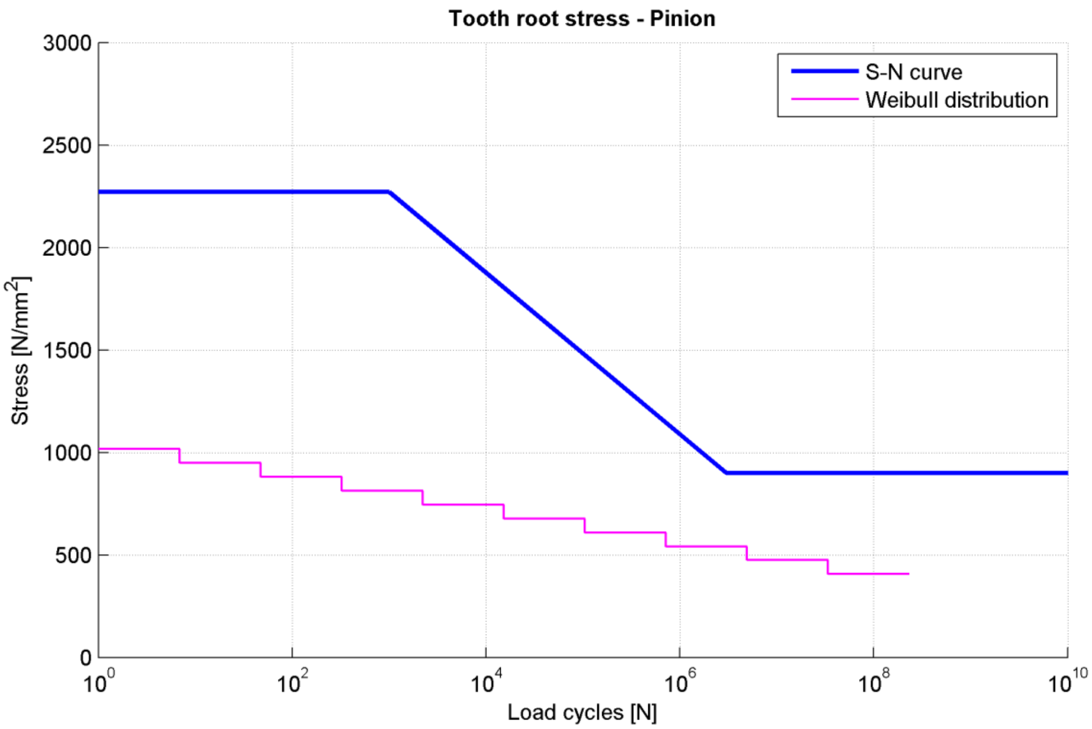


Figure 6. *Example of discrete Weibull distribution on contact stresses.*



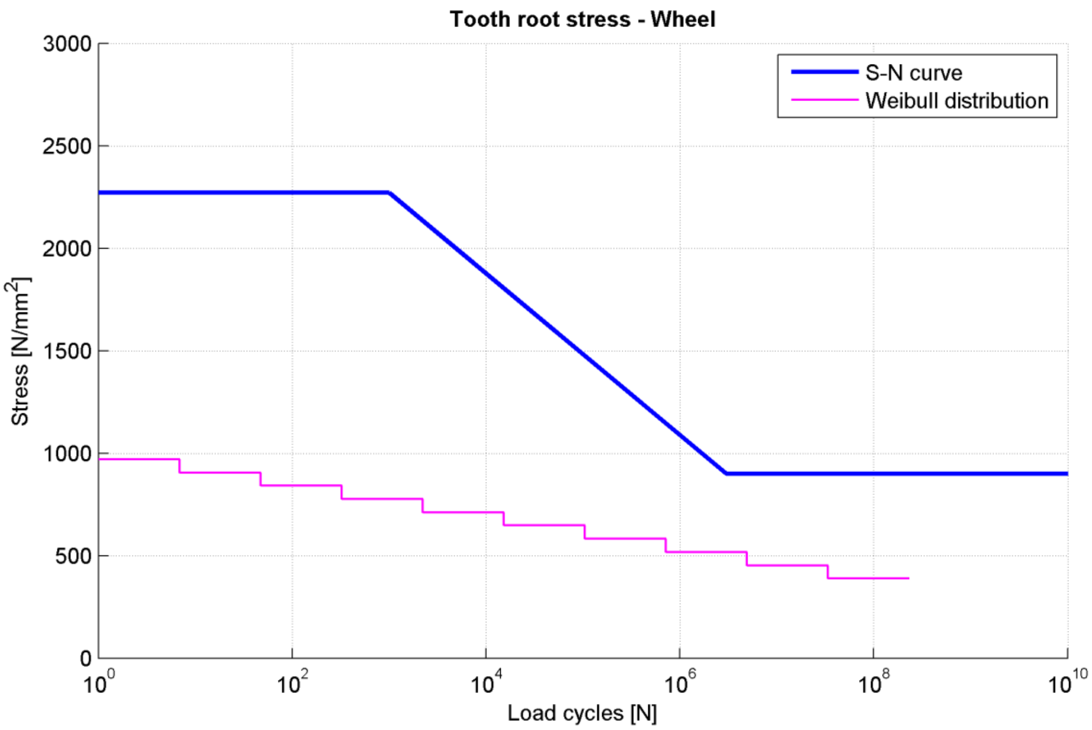


Figure 7. *Example of Weibull distribution for tooth root stresses.*

As a Matlab function output we get all the calculated factors used in calculation as well as the most important part for the preliminary analysis, safety factors. All of these are set in a structured format so they would be easily available. The results could also be further implemented in the actual classification process if the needed output information is saved into some output file.

All of the needed gear data was written into a separate m-file which enables to use it also in the DIN based gear calculation. Both of these calculations have some of their own values taken from tables but the geometry and lubrication features are the same in both cases.

5.3.2 DIN gear rating

All the other gear standards are similar because the component has the same mechanical properties but there are slight differences in formulations and therefore in the results. DNV rules being much more recent have much more sophisticated way to calculate scuffing than the older DIN rules. But the biggest differences between the DNV classification notes 41.2 (2012) and DIN 3991 (1988) standard are in the way the ice load spectrum is considered. Also the DIN standard doesn't evaluate subsurface fatigue which is one of the most restricting features in the DNV rules.

The ice load spectrum with DIN 3991 is calculated using ISO 3663-6 (2006) standard. The approach differs from DNV in a way that it is done by assessing the accumulated damage on higher torque that is equivalent on a lower torque rate and higher number of equivalent load cycles. This means that the damage caused by higher torque can be reduced to a lower torque with equal damage. The number of load cycles increases so that the equivalency can be achieved. This is done for both pitting and tooth root damage until Wöhler-damage line knuckle point is met. (ISO 2006, p. 10-14) At the knuckle point equivalent torque is reached and application factor can be calculated which takes into account all the accumulated damage. The load spectrum with equivalent load cycles is shown in Figure 8.

The used ice load spectrum is similar two-parameter Weibull distributed spectrum as in DNV calculation. The difference is that this spectrum is created using the continuous distribution and it is divided into discrete points with torque amplitude value and corresponding amount of cumulated load cycles. In other words, equivalent number of cumulated load cycles is calculated on a discrete torque load based on the damage that the higher torque loads cause and it is calculated for both tooth root and pitting cases. From the standard we get the specified amount of load cycles (separately for both cases) when the knuckle point is reached and compare equivalent load cycles to those. Loads below the knuckle point can be ignored. When the value is reached the equivalent torque is interpolated between the two closest points and as a result we get the equivalent torque exactly at the knuckle point. From the figure we can see that in this example case tooth root gives higher torque and therefore defines equivalent application factor. It is used in the gear calculation afterwards.

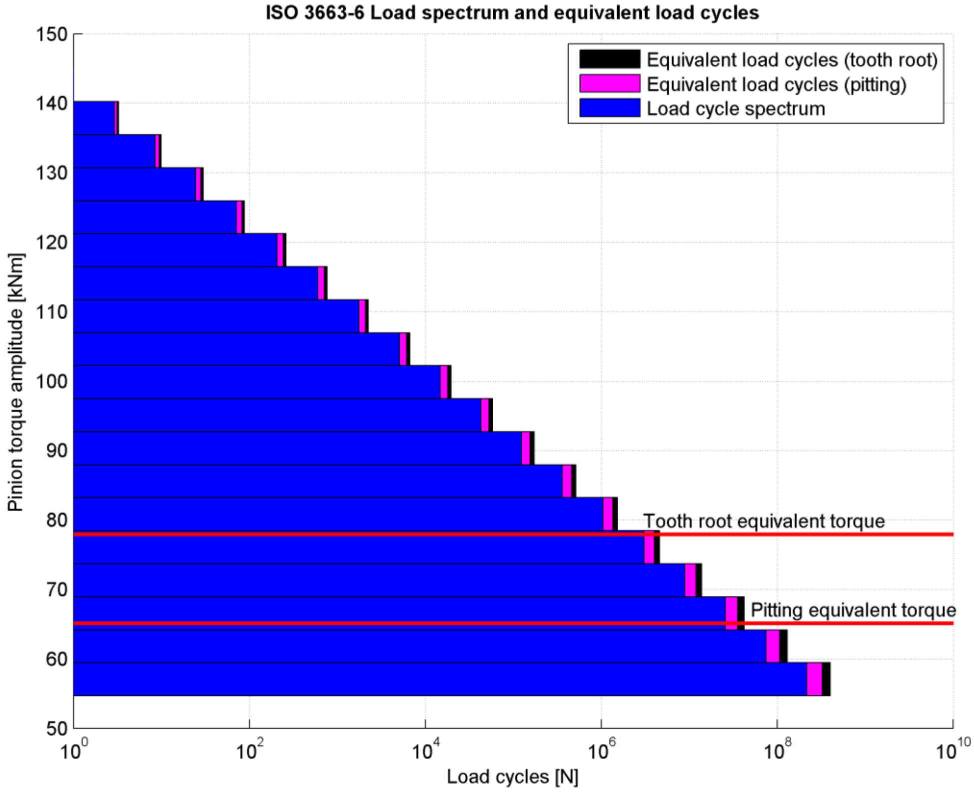


Figure 8. *Example of equivalent torque spectrum.*

This load cycle spectrum approach creates difference in the resulting power between the two gear calculations. In DNV rules the safety factors are calculated in each load bin independently. In this approach we need to do the gear calculation only once for the higher equivalent application factor and this calculation is done for infinite gear life. Scuffing is however not taken into account in the equivalent application factor so it needs to be calculated with the highest load separately, like in the DNV gear rating.

In the gear calculation the safety factors are calculated for contact stresses, tooth root stresses and scuffing. The idea is to prevent all the same kind of damages as in the DNV gear rules and all of these cases have their own requirements for safety.

The required safety factors differ in each classification society that uses DIN gear calculation procedure so comparison of the results is not that simple. This however shouldn’t be problem since customer wants the vessel and propulsion machinery to be classified according to certain classification society’s rules and ice class. Still, this fact also means that the same results can’t be applied for all the societies and the calculations need to be done for each society independently in order to be as competitive as possible.

This DIN gear calculation process is applied as a Matlab function and the results are similarly arranged as structured outputs like in the DNV gear calculation. This should make the outputs also easier to understand and use in the final program. It also facilitates the utilization of the Matlab generated output files in Excel or in other formats.

5.4 New process for the calculation tool

The goal is to make an effective tool for sales team that's easy to approach and use. The current process in use could be adapted to this by reducing the amount of input variables to a set of initial input parameters that are needed and doing all calculations based on those in a single program. Although the initial parameters can't all be chosen accurately and some amount of estimation is still needed, it would simplify and speed up the process. Initial parameters that are at least needed: input power and rotational speed, ice class, gear data, propeller data, type of main machinery and a reference project which is used as a basis for sales calculation.

Reference project could be pre-chosen from one of the old projects because lumped mass data stays relatively unchanged for most of the shaft line components in all of the projects. The most important components that need to be considered are power unit and flexible coupling which may be externally bought components for the propeller manufacturer. Propeller design also varies so the end user needs to be able to change these parameters if necessary.

The idea for improved process is shown in Figure 9 and it is meant for a single thruster unit. The main idea is that the user doesn't need to access anything else than the user interface and that the correct data for gears and simulation is chosen automatically. This is done by setting all the gear data and a pre-chosen simulation model in to the program and those are chosen with a couple of parameters given in the user interface. The correct gear data is chosen with tooth numbers and the simulation model is chosen to be either electric motor or diesel engine model. Then the model is simulated and the correct gear rating function is called based on the inputs.

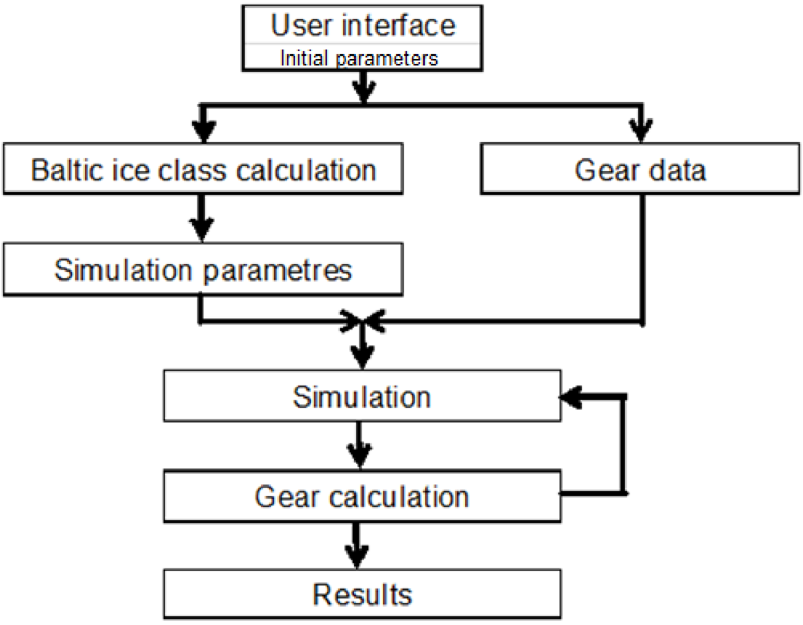


Figure 9. *Basis of automated calculation process.*

Other parameters that are given in the user interface are ice class, power, rotational speed and the propeller data needed for the calculation of maximum ice induced torque, equations (1)-(4). Some additional parameters are also available for adjusting the power unit output values like the diesel engine design torque margin and the amount of allowed over torque for electric motor. Simulation time and the simulated ice load cases can be chosen as well.

This automated process for one thruster unit calculates ice loads and amount of load cycles with the given propeller data and ice class. After that it uses the correct simulation model for evaluation of maximum loads for the gears and the gear data that is to be used. This is similar for all the of classification societies using Finnish-Swedish ice class rules. The gear calculation is then done by using those maximum loads and according to the chosen gear rating rules. As a result user would see the safety factors compared against the required safety factors for gears and the iterated power along with relevant figures from gear calculation and simulation.

This described process in Figure 9 was done with Matlab during this study using the simulation model and gear calculation presented earlier. The basic idea of the program can be seen in Figure 10 and it was done for the Finnish-Swedish ice class rules. Option was left to add polar class calculation later.

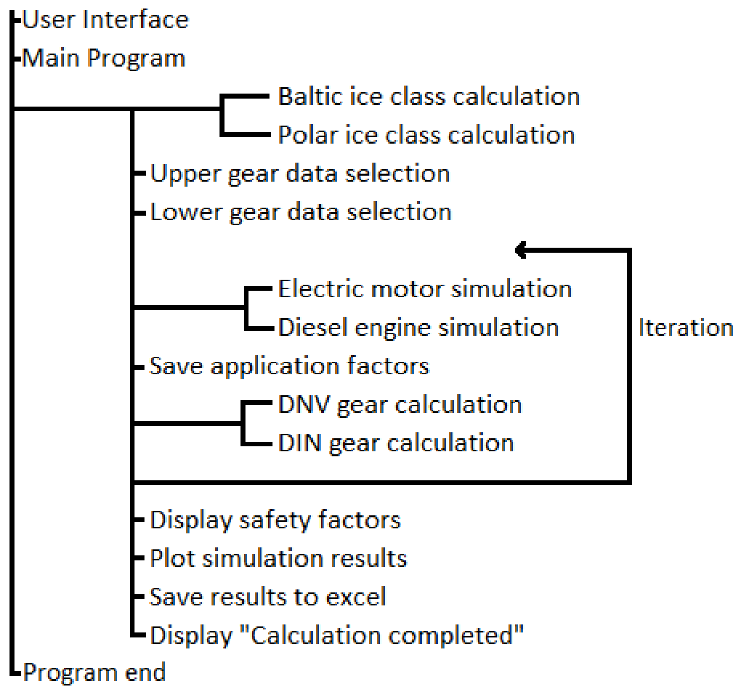


Figure 10. *Flowchart for the calculation program.*

This program was done by setting all data needed for simulation into one “user interface” type of m-file and all the gear data’s (several gear pairs due to different gear ratios) on separate m-files. The user interface calls for the main program which in turn calls for calculation procedures and simulation in the necessary order. Simulation models were made for both electric motor and diesel engine unit independently and the gear calculations are made as Matlab functions. Because all of the calculations are on separate files, it should be relatively easy to find and update all necessary parameters or equations on this calculation process if needed.

The automated iteration process was not implemented during this study and the power needs to be adjusted manually. The reason for this is that this way the program can be used in optimization of some other characteristic as well. The power adjustment means that when the resulting safety factors are too low for gears, nominal power for the thruster unit needs to be lowered and the program needs to be run again. But just by having all the calculations automated and in the same program speeds up the process tremendously. An option to add automatic iteration later was commented in the Matlab script.

The simulation results for upper gear, lower gear and flexible coupling are printed in figures and saved automatically in png format. These figures show the changes in rotational speed, torque and application factor during the simulation. Example of the resulting figures is presented in appendix 3. Additionally required minimum and the calculated safety factors are shown for both upper and lower gears. Also text telling if the gears are valid or not, is displayed.

With Matlab it’s relatively simple to save to results in excel format as well and this was done for the part of simulation parameters, simulation results and gear calculation safety factors. By adding all the other calculated gear calculation results this could also be used for classification purposes.

Total time for one calculation without iteration is 20-30 seconds in real time depending on the length of the simulation time. This is relatively good but it can be even faster without saving and plotting figures from results. Those are however vital for checking the correctness of the simulation results. Also the calculation needs to be done 5-6 times on average which means that the whole process takes around 3 minutes. This should be quite okay considering that there is a complex simulation done multiple times in that overall real time as well.

This program makes the power evaluation easier but there is still the question about how to choose a suitable reference project and components when doing a sales offer. There can be quite significant differences in the power unit, flexible coupling and propeller properties from project to project and these affect the end results. Sensitivity analysis is a good way to approximate the effect of these.

6. SENSITIVITY ANALYSIS

Usually in unit sales offer stage known factors are: power and ice class. Power is required by customer either directly or indirectly as desired thrust and the ice class requirement is based on ships operational profile. The profile is also used as a basis for the propeller design which helps to narrow down options a little bit. These variables however do not really limit the unit choice all that much and the best fitting unit needs to be estimated and chosen from the unit selection. After that all the needed components and parameters need to be chosen for that unit. This requires very much experience so that every choice is done correctly for preliminary sales calculations. Sometimes there may be other important variables available, like propeller depth or input speed, which may help to limit thruster unit choices.

After the unit is chosen, the newly developed tool is put to use. Gears and shafts are usually known and propeller manufacturers would like to use the same designs for as many projects as possible in order to make things more straightforward. This is also a good thing regarding ice class calculation but this still basically means that the offered propeller design and mass elastic data for the power transmission needs to be estimated. But using experience and data from previous designs is recommendable as it gives better starting point. Moreover the shaft and gear inertias and stiffness’s do not vary that much between projects.

Some components like engine or motor and flexible coupling still vary from project to project and sensitivity analysis is needed to study how these choices affect the allowed power of the thruster unit. Electric motors are able to provide over torque for short periods of time so this effect is also studied. This also helps to narrow down choices that have to be made at later stages of the project design.

The choice of power unit is also important because it also defines dynamic characteristics, the propulsion systems input speed and it has an effect on the choice of flexible coupling. Propeller also has high amount of influence but this can be more impacted by the designer. These are studied next with the developed program and with the goal of maximizing power output in an ice classified thruster.

6.1 Calculation parameters

The chosen azimuthing propeller unit that is taken into closer inspection is Rolls-Royce US 305 and the datasheets for it can be seen in appendix 1 and 2. In the datasheets it can be seen that the unit can be driven with either diesel engine or electric motor and it can have either CP or FP propeller. The propeller design can furthermore be open or ducted.

The propeller unit used in this study has gear sets for 750, 800, 900, 1000, 1200 and 1600 rpm nominal input speeds and the nominal power in open water for these is around 3000 kW. In other units rpm ratings of 1500 and 1800 are also common but they're not available for this unit at this time. These set some limits to this study and the selection what kind of power units can be used.

The data used in the calculations is presented in Table 6 and unless certain component or parameter is inspected more carefully, the parameters are kept constant during the calculations. For example when the effect of diesel engine is studied coupling and propeller are kept the same.

Table 6. *General data for the sensitivity analysis.*

Component	Description	
Power unit	Diesel engine MAK 9M25C Power: 3000 kW Nominal speed: 750 Design margin: 15 % Total mass moment of inertia: 435 kgm^2	Electric motor ABB AMI 560L6L BAFTMH Power: 3000 kW Nominal speed: 1200 Allowed over torque: 10% Total mass moment of inertia: 123 kgm^2
Coupling	Vulkan Rato-R G312TR Stiffness: 554.02 kNm/rad Relative damping ($\eta=\psi/2\pi$): 0.180	Centa CL-75-FF1-CX-78-60 Stiffness: 218,04 kNm/rad Relative damping ($\eta=\psi/2\pi$): 0.165
Propeller	CP, open, wing propeller Diameter: 2.8 m No. of blades: 4 Pitch at 0.7r: 1.259*D Mass moment of inertia in water: 1721 kgm^2 Propeller depth: 2.8 m	FP, ducted, wing propeller Diameter: 3.2 m No. of blades: 4 Pitch at 0.7r: 0.861*D Mass moment of inertia in water: 2380 kgm^2 Propeller depth: 3.2 m
Gear calculation	DNV CN 41.2 (2012) Applied load cycles: result of equation 7 to all gears Amount of load bins: 10	
Lower gear	Ratio: 3.5 (14-49)	
Upper gears	Ratio: 1.038 (26-27, 750 rpm) 1.125 (24-27, 800 rpm) 1.217 (23-28, 900 rpm) 1.375 (24-33, 1000 rpm) 1.652 (23-38, 1200 rpm) 2.238 (21-47, 1600 rpm)	

CP propeller is chosen with the diesel engine as is the more common practice and FP propeller is chosen with the electric motor. Lower gear is the same for each input speed and as the propeller shaft speed is wanted to be kept relatively the same for all input speeds, upper gear ratios change. This way the propeller blade tip speed stays on the optimum range.

In gear calculation the amount of applied load cycles is set the same for both wheel and the pinion and for the upper and lower gear. This is a little bit inaccurate and conservative as the lower gear pinion teeth should be the ones experiencing the most load cycles and lower amount could be used for upper gears. The amount of load cycles is very much dependent on the propeller centerline depth which is chosen to be same as the propeller diameter.

DNV marine gear rating rules are used because those seem to restrict output power much more. Therefore it is easier to see how the inspected characteristics behave because the results have much bigger influence. This is shown later on when the best and the worst cases for DNV gear results are compared with the DIN standard gear calculation results in chapter 6.7.

In Table 7 it’s shown why low amount of load bins is chosen for the calculation. The powers are iterated for ice class 1A super with 1 kW accuracy so that the safety factors are minimized but acceptable. It should be noted that because the gear calculation load spectrum is done with discrete frequency spectrum and the amount of load bins is not to be less than 10 (DNV 2011, p. 60).

Table 7. *Effect of the amount of load bins in the gear calculation.*

Amount of load bins	Iterated power, electric [kW]	Iterated power, diesel [kW]
10	2740	2652
15	2824	2750
20	2873	2797
25	2900	2826
50	2901	2826
100	2928	2855
1000	2964	2874

The resulting power is highly influenced by the general system dynamics but as it can be seen, higher amount of load bins leads to higher allowed power. The reason for this is that the stress distribution is done with stress histograms and the fewer load bins are used, the more conservative the results are (DNV 2011, p. 61). This of course has great effect on the iterated powers as well. With higher number of load bins the load spectrum approaches the continuous Weibull distribution. This effect can be seen in Figure 11 where the actual calculation points are the corner points above the continuous spectrum.

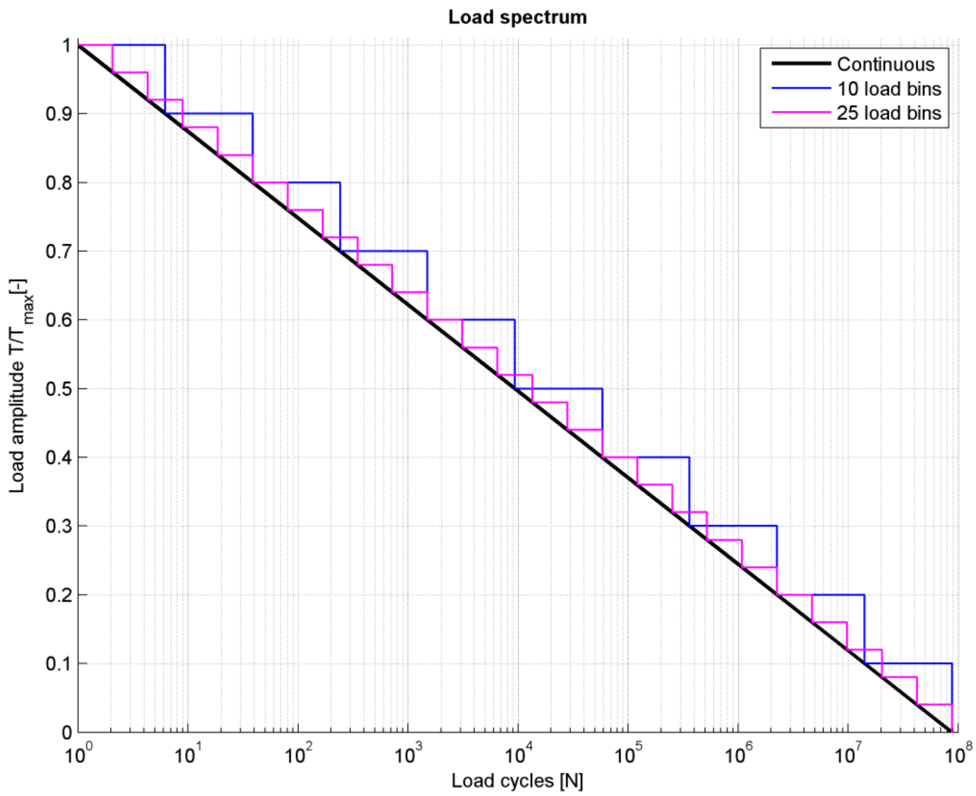


Figure 11. *Amount of load bins on the ice load distribution.*

The resulting safety factors also seem much more consistent in the iteration when there are more discrete steps. Higher amount of load bins of course also slows down the calculation process because the calculation needs to be done many more times but the gear calculation itself is not that demanding process with Matlab, so the calculation time doesn't change much.

The effect of load bins doesn't have that much meaning if the restricting factor doesn't happen to be the safety factor for subsurface fatigue. Subsurface fatigue is calculated for all the load bins independently and because the lower amount of load bins is more conservative, it means that the safety factors are lower. In these cases different amount of load bins affects the results but if the power is restricted by scuffing, lowering the nominal power is the only way that will help if the lubrication is not changed. This is because the scuffing is calculated only with the maximum loading condition and it is not affected by the amount of load cycles or load bins.

During preliminary calculations for sales offer the effect of load bins could be used as a more conservative estimation for the offered units. Using 10 load bins would take many uncertainties into account and using more load bins in the design phase would give better, less conservative results for the design. In this study lower amount of load bins is chosen because it's easier to see the studied effects. This can be seen for example with the diesel engine inertia analysis.

6.2 Diesel engine

Diesel engines have many differences in the torsional vibration calculation because between two diesel engines the amount of cylinders and crankshaft, flywheel and dampener properties may vary in the same power range and with the same output speed. This gets even more complicated as the output speeds also vary. Diesel engines used in calculations are shown in Table 8 and these are chosen based on the engine output speeds and available gears.

Table 8. *Diesel engine output and mass moment of inertia.*

Engine	Maximum continuous power [kW]	Nominal speed [rpm]	Mass moment of inertia [kgm^2]	Datasheet
MAN 9L27/38	2970	750	510	L27/38 Project Guide – Marine
MAK 9M25C	3000	750	435	M25 C propulsion project guide
Wärtsilä 6L34DF	3000	750	440…490	Wärtsilä 34DF Project Guide
Wärtsilä 6L32	3000	750	500…560	Wärtsilä 32 Project Guide
MAN 6L32/40	3000	750	527	L32/40 Project Guide
MAN 6L35/44DF	3180	750	680	L35/44DF Project Guide – Marine
Wärtsilä 9L26	2925	900	183	Wärtsilä 26 Project Guide
Wärtsilä 9L26	3060	1000	183	Wärtsilä 26 Project Guide

Although the engines consist of multiple rotating masses, the usual way they are presented in manufacturer catalogues is with total mass moment of inertia. The more detailed mass-elastic models are used in torsional vibration calculations which can better take into account the engine damping but these models are considered proprietary information. Reducing the engine to one total mass moment of inertia shouldn't however have huge impact on the results and the simplification is also suggested in the DNV rules. The effect of damping of the engine is also low so it can be neglected. (DNV 2011, p. 43) This was also studied earlier with one example and confirmed with the simulation model.

Furthermore this program is only thought to be used in sales offers which are only preliminary calculations. So everything can't be taken into account and it's not worth it to calculate with extreme precision because the values will most definitely change in the design phase. Total mass moment of inertia data for different engines is also much easier to compare as those are much more easily available.

The total mass moments of inertia presented in Table 8 contain the flywheel inertia and the values are the minimum possible inertias for the engines. Already before doing any simulations it can be seen that the difference in the inertias is quite high. The lowest inertia is over three times smaller than the highest inertia in the table. If we take into account that those engines have different nominal speeds and compare only similar speed and power engines, the highest inertia is 56.3% higher than the lowest. That seems more reasonable but it is still a very significant difference considering that both of these engines are designed for propulsion applications.

Even higher engine inertias are still plausible because the engines are usually tailored for each system independently. The mass moments of inertia in Table 8 are the minimum possible inertias but those can be easily increased because of a different, heavier flywheel for example. This is the case when the engines are customized so that the best possible performance can be assured and bad dynamic behavior can be avoided. But this makes estimation in sales offers harder if the customer doesn't already have some plans

Engines are chosen to have reserve torque by the margin of 15% more than propeller torque curve. This is because most of the engines are supposed to be rated for 85% of the full load at nominal speed for sea-going ships (M25 C Project Guide, 2012, p. 6; Wärtsilä 32 Project Guide, 2014, p. 6; Wärtsilä 26 Project Guide, 2014, p. 6). The 15% reserve is left to take into account for example different weather conditions.

Engine manufacturers usually provide their own characteristic curves as well for their engines but their shape is usually similar to propeller torque law. An example of the engine cylinder output is shown in Figure 12 and it is used as a reference for the sales offer engine output modeling. The speed above nominal rotating speed is not modeled because the engine speed is not allowed to go above that in the simulation model.

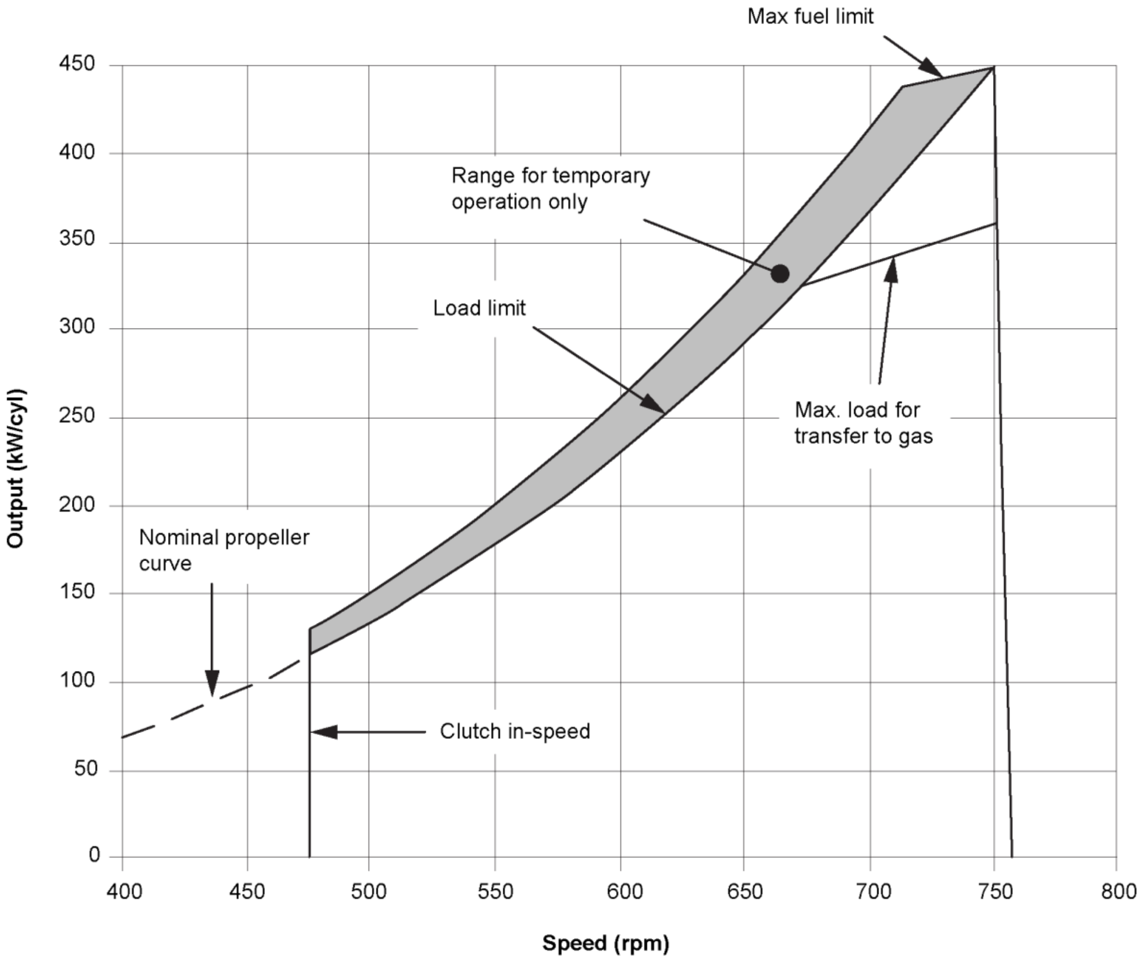


Figure 12. *Engine output for one cylinder with CP propeller (Wärtsilä 34DF Project Guide, 2014, p. 6).*

The engine output performance curve for the preliminary calculations is shown in Figure 13. The reserve of 15 % is left by default because of torque peaks but this could be adjusted in the program interface if needed. This curve is also just a crude approximation for the sake of simplicity in the preliminary calculations and it is thought to be a simple way to be applied in all of the available nominal rotational speeds.

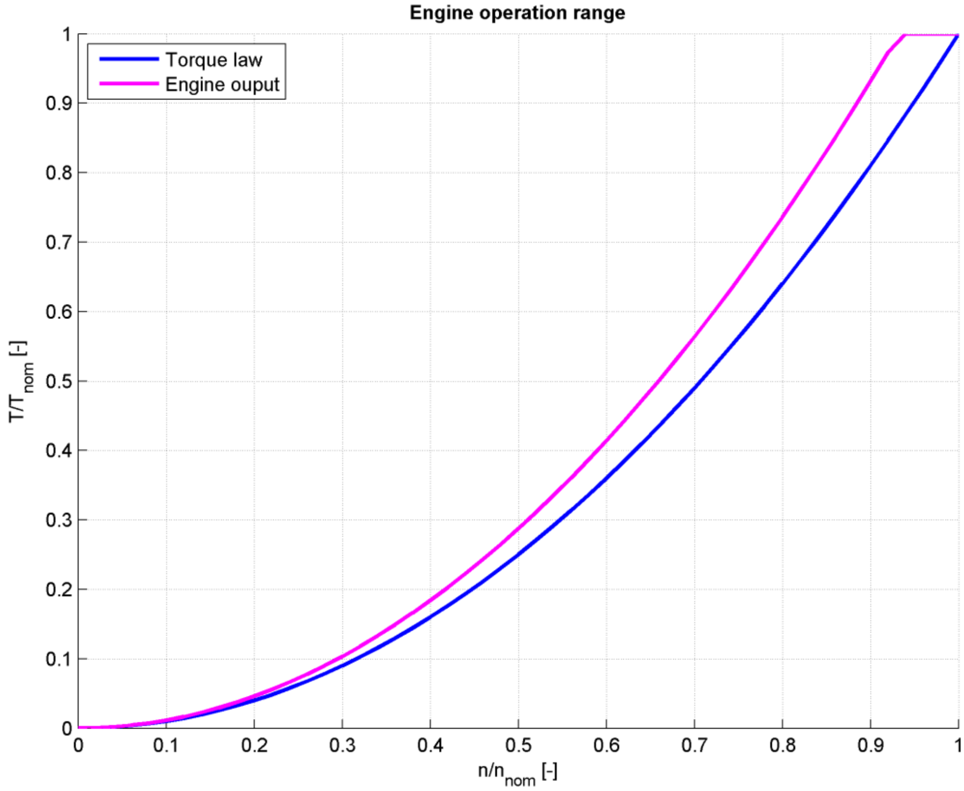


Figure 13. *Approximation of engine torque – speed curve.*

The presented crude approximation is of course inaccurate at lower engine speeds but it is described like that for numerical reasons. Additionally the ice loads occur in MCR conditions so the accuracy on the lower speeds shouldn't have that big of an effect. In reality the engine would stall much earlier and it has to be taken into account later in the design phase. The engine also needs to achieve certain rotational speed before coupling is engaged. This operation is however disregarded in this study because the ice loads at MCR condition are restricting the power output and therefore much more of an interest in this study.

This kind of a curve can be automated quite easily with a couple of equations in the model so this is easy to use in the sales calculations. Because engines have slightly different characteristics in the final design, it would be preferable to use proper output characteristics as well as the more accurate mass elastic model for the engines in the final design.

The effect of engine inertia was studied with the described performance curve and the diesel engines in Table 8. As a result, we get the maximum allowed power with the minimum safety factors for gears and those are presented in Table 9. Calculation is done for the engines that have a range of inertia in the table for both lower and upper limits.

Table 9. *Results for diesel engines.*

Engine	Nominal power [kW]	Iterated results [kW] ICE-1C	ICE-1B	ICE-1A	ICE-1A super
750 rpm engines					
MAN 9L27/38 (510 kgm^2)	2970	2970	2970	2683*	2614*
MAK 9M25C (435 kgm^2)	3000	3000	3000	2716*	2652*
W6L34DF (440 kgm^2)	3000	3000	3000	2712*	2652*
W6L34DF (490 kgm^2)	3000	3000	3000	2691*	2629*
W6L32 (500 kgm^2)	3000	3000	3000	2690*	2926*
MAN 6L32/40 (527 kgm^2)	3000	3000	3000	2676*	2613*
W6L32 (560 kgm^2)	3000	3000	3000	2662	2600*
MAN 6L35/44DF (680kgm^2)	3180	3000	3000	2624	2561*
Other engines					
W9L26 (900 rpm)	2925	2823	2748	2397*	2344*
W9L26 (1000 rpm)	3060	3000	3000	2733*	2674*

*) Engine stalls or the rpm drops below 40 % of the nominal

In all of the cases maximum power of 3000 kW is determined by the gears in the free running condition and the iteration is done with 1 kW accuracy. The calculations were done for each ice class and the worst and the best results were highlighted. We can see that because the ice torque loads are higher for higher ice classes, those also have lower maximum allowed power. The power drop on 750 rpm engines seems to be around 3.4% with 56.3 % increase in inertia.

It also seems that subsurface fatigue is a common restrictive phenomenon in all of the cases. In DNV rules this is very much dependent on the depth of surface hardening and the load spectrum. But because the 750 rpm engines all have the same gears, we can see that higher inertia results in a lower power. These results are plotted in Figure 14 with some additional points in order to better understand the shape of this phenomenon.

Input speed of 1000 rpm seems to be in the same range as the 750 rpm engines but because the 1000 rpm engine has much lower inertia, the power is higher than in any of the other results. That may also be because of the better gear design for this use. The 900 rpm case seems to have gears restricting the output power even on the lower ice classes so this would suggest that the gear design is not optimal for this kind of use. The effect of system input speed is studied more later on in chapter 6.4.

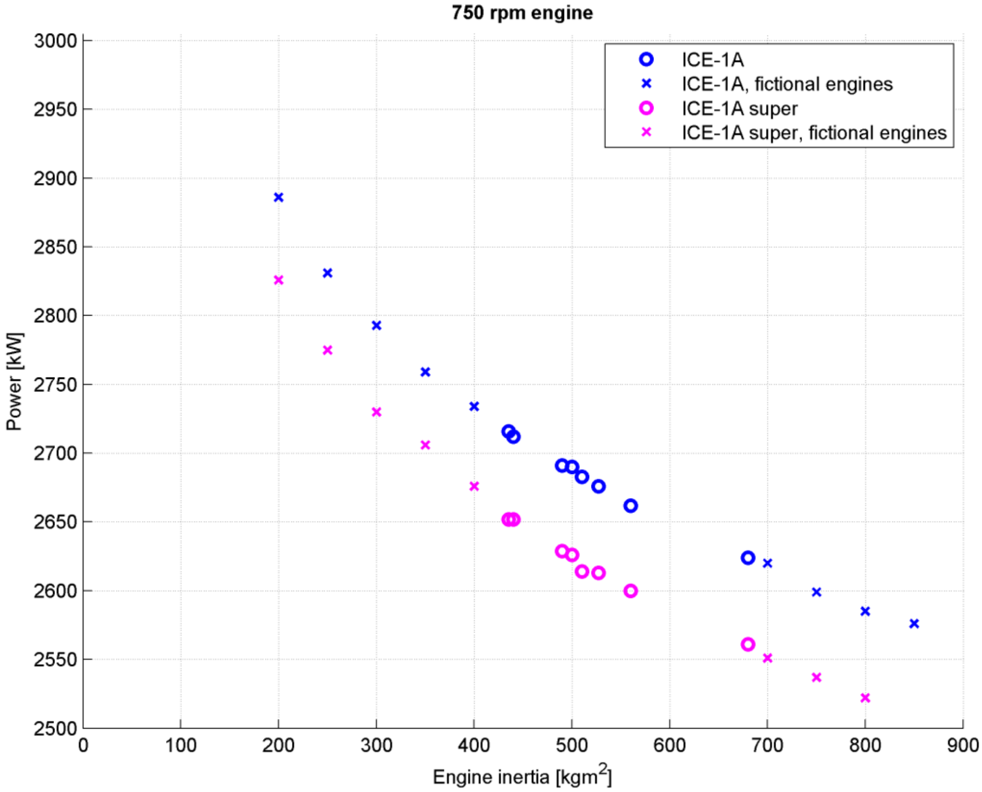


Figure 14. *Effect of engine inertia.*

The results in Figure 14 show that for the 750 rpm engines higher inertia causes higher loads and therefore lower power. Higher inertia is also known to resist changes in rotational speed and because the engine output performance is described with rotational speed – torque curve, those two influence on each other. Smaller speed drop means that the torque load is higher and the ice torque load is added to that. Ice torque load is mostly dependent of the propeller design and doesn't change much so the total maximum is higher with lower speed drop.

With these 750 rpm engines we don't see any adverse dynamic characteristics appearing on the Figure 14. This is mostly not the case because as there may be some points where ice loads excite some natural frequencies of the system and cause higher load peaks. This would result in a lower power and it would cause discrepancies in the results. This case would seem to have quite optimal choices for components for this rotational speed.

The highlighted best and the worst cases in Table 9 for 750 rpm are shown in Figure 15 for ice class 1A where the speed drop effect can be seen most clearly. With 1A super ice class engines seemed to stall in many cases. The order of ice load cases is changed from the one presented in Table 3 to order 1-3-2 because the load case 2 seems to be the most demanding and can stall the engine. With this order of loads it was possible to simulate all of the load cases in one simulation run even if the engine stalls because the most demanding load case is last.

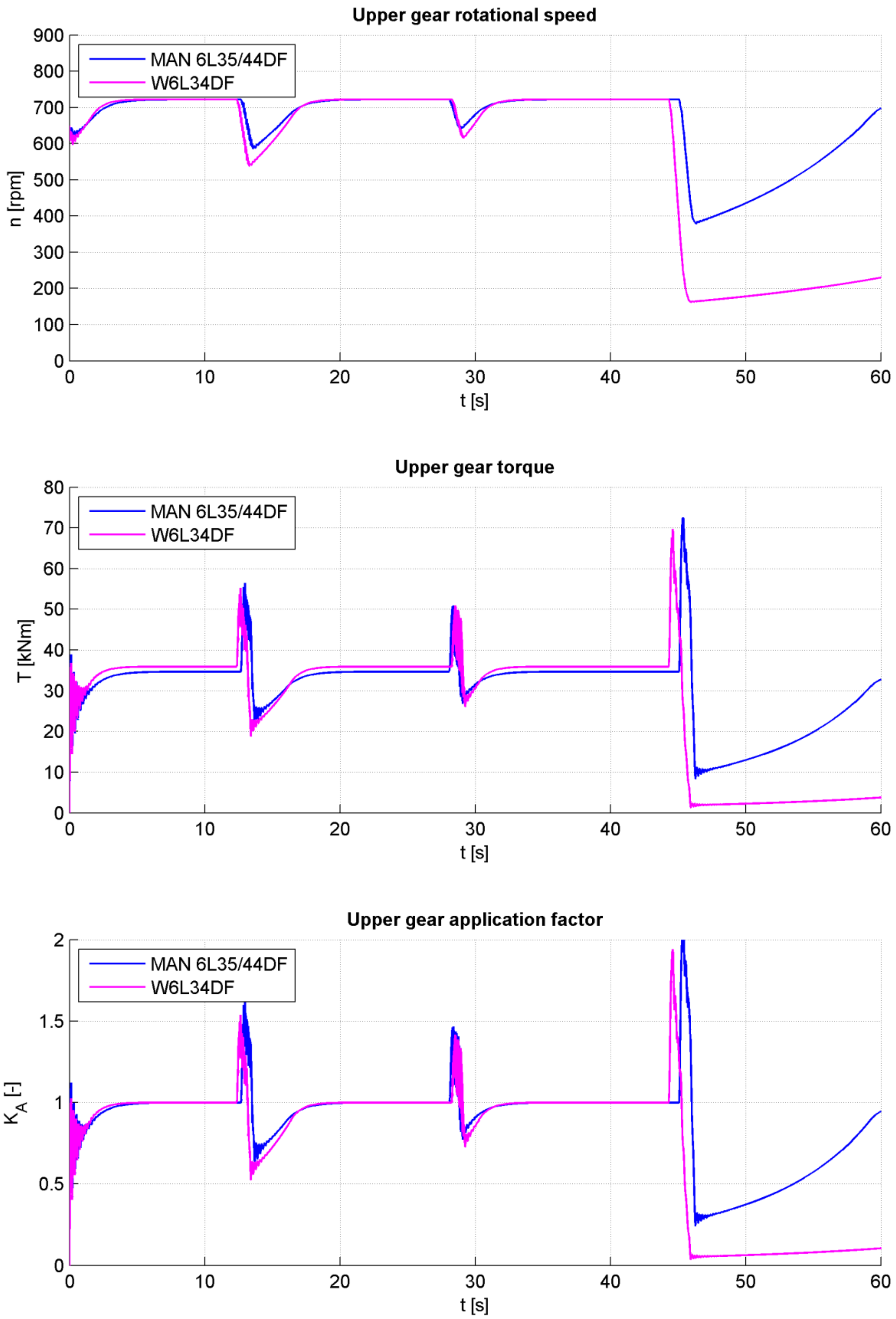


Figure 15. *Comparison of the best and the worst performing 750 rpm engine.*

The engine inertia doesn’t seem to have as big of an effect to the results as the number of load bins so the variance in the engine inertias should be accounted for. Low number of load bins should be therefore used in the sales offer calculations. In the design phase it needs to be thought out which engine is the best because of the stalling issues and the choice is not that simple. Bigger problem seems to be that the engine inertia can change with the output speed quite drastically and that the gear design can affect the results even more. This is even harder because there wasn’t that many engines available with this power for propulsion use and any kind of trend for inertia with different output speeds can’t be assumed. All that can be said is that all of the sales offer cases need to be calculated independently because of the differences in the gears.

All of these engines have cylinders inline so the V-engine can have different kind of properties, especially if those are modeled with engine manufacturer’s TVA data. It should still be somewhat similar with the inline engines but the output performance needs to be confirmed.

6.3 Electric motor

In electric motors there is a lot less variables in the mass elastic data when comparing to diesel engines but electric motor inertia may still vary in a similar manner as diesel engines total mass moment of inertia. Furthermore the electric motor performance is a little bit different and enables the use of over torque for example. The effect of these properties is studied here.

The motors in Table 10 are catalogue motors from the electric motor manufacturers. These values only give estimation on what range the inertia is because these motors are usually modified for each system by the manufacturer. All data for ABB motors is from High voltage induction motors Technical catalog (2011) and data for WEG motors is from their website Electric Motor Selection – M-Line –MGF low and high voltage motors (2015).

Table 10. *Electric motor output and mass moment of inertia.*

Electric motor	Maximum continuous power [kW]	Nominal speed [rpm]	Mass moment of inertia [kgm^2]
750 rpm motors			
ABB AMI 630L8A B	3150	744	315.0
ABB AMI 630L8A B	3150	745	330.6
WEG MGF 10408	3150	746	352.7
ABB AMI 630L8A B	3150	746	378.6
WEG MGF 710A	3150	746	415.1
WEG MGF 710A	3150	746	457.9
900 rpm motors			
WEG MGF 10408	3150	895	313.8
WEG MGF 630B	3150	895	332.7
WEG MGF 12007	3150	895	405.9
1000 rpm motors			
WEG MGF 8808	3150	994	126.5
ABB AMI 560L6A B	3150	994	163.5
ABB AMI 560L6A B	3150	994	172.9
ABB AMI 630L6A B	3150	993	236.0
WEG MGF 10408	3150	994	305.0
WEG MGF 710A	3150	994	361.4
1200 rpm motors			
ABB AMI 560L6L BAFTMH	3000	1200.5	123.0
WEG MGF 560B	3150	1193	123.7
WEG MGF 560B	3150	1194	127.1
WEG MGF 10408	3150	1194	234.7

Although the inertias seem to be quite consistent for each rotational speed, there are significant differences. The biggest differences are in the 1000 rpm motors as the highest inertia is 185% higher than the smallest, with 750 rpm motors the difference is much smaller but still significant. It is also noteworthy that with the same number of poles in the rotor, motors seem to have inertias in the similar range. The difference in those motors is that they operate on a different frequency, 50 or 60 Hz. This can especially be seen in 750 and 900 rpm motor inertias.

The difference in inertias can be explained with different structures, pole numbers and input voltages. These motors are totally enclosed air-to-air cooled motors with input voltage range from 3000 V to 13800 V. Water-to-air cooling is commonly used onboard but this shouldn't have much effect on the inertia.

Simulation was done for each motor in order to find out how the inertia affects the maximum output power. All the results are in Table 11 and those are calculated with 10% allowed over torque.

Table 11. *Results for electric motors.*

		Iterated results [kW]			
Electric motor	**Nominal power [kW]**	**ICE-1C**	**ICE-1B**	**ICE-1A**	**ICE-1A super**
750 rpm motors					
ABB AMI 630L8A B (315.0 kgm^2)	3150	3000	3000	2854	2700
ABB AMI 630L8A B (330.6 kgm^2)	3150	3000	3000	2836	2693
WEG MGF 10408 (352.7 kgm^2)	3150	3000	3000	2825	2667
ABB AMI 630L8A B (378.6 kgm^2)	3150	3000	3000	2806	2645
WEG MGF 710A (415.1 kgm^2)	3150	3000	3000	2779	2624
WEG MGF 710A (457.9 kgm^2)	3150	3000	3000	2760	2595
900 rpm motors					
WEG MGF 10408 (313.8 kgm^2)	3150	2797	2722	2357	2113
WEG MGF 630B (332.7 kgm^2)	3150	2792	2714	2341	2037
WEG MGF 12007 (405.9 kgm^2)	3150	2771	2685	2292	1850
1000 rpm motors					
WEG MGF 8808 (126.5 kgm^2)	3150	3000	3000	2955	2825
ABB AMI 560L6A B (163.5 kgm^2)	3150	3000	3000	2887	2725
ABB AMI 560L6A B (172.9 kgm^2)	3150	3000	3000	2873	2708
ABB AMI 630L6A B (236.0 kgm^2)	3150	3000	3000	2789	2619
WEG MGF 10408 (305.0 kgm^2)	3150	3000	3000	2726	2532
WEG MGF 710A (361.4 kgm^2)	3150	3000	3000	2666	2475
1200 rpm motors					
ABB AMI 560L6L B (123.0 kgm^2)	3000	3000	3000	2868	2711
WEG MGF 560B (123.7 kgm^2)	3150	3000	3000	2870	2714
WEG MGF 560B (127.1 kgm^2)	3150	3000	3000	2855	2703
WEG MGF 10408 (234.7 kgm^2)	3150	3000	3000	2693	2516

The effect of inertia seems to be similar as with diesel engines, higher inertias seem to cause higher loads. With the biggest inertia difference motors, 1000 rpm, we can see 12.4% power drop when comparing to the best case. The power drop can be considered quite big but because of the 185% change in inertia, this doesn't seem to be the most defining factor.

With these motors we can see better that there must be some gear design issues with the 900 rpm speed. All of the other rotational speed cases are quite close to each other which confirms the findings already done with the diesel engines. Gear designs limit the allowable power the most. This is even worse in this case as the limiting factor for the 900 rpm upper gears is scuffing which means that not even the amount of load bins will change the results. For other input speeds the limiting factor is the subsurface fatigue so the differences could be mostly taken into account with the amount load bins in the gear calculation.

Allowing electric motors to have over torque for limited time smoothens out vibrations in thrust (Riska 2011, p. 14-15) and this is a useful feature in ice navigation. Effect of the amount of over torque on the power drive dynamics and the allowed power is studied next. This is done with a couple of torque rate of increase limits and over torques ranging from 0-50 %. These results can be seen in Table 12.

Table 12. *Results for the effect of over torque.*

		Iterated results [kW]			
Over torque [%]	**Nominal power [kW]**	**ICE-1C**	**ICE-1B**	**ICE-1A**	**ICE-1A super**
50 %/s torque rate of increase					
0	3000	3000	3000	2886	2740
10	3000	3000	3000	2868	2711
20	3000	3000	3000	2872	2712
30	3000	3000	3000	2869	2704
40	3000	3000	3000	2865	2717
50	3000	3000	3000	2872	2720
100 %/s torque rate of increase					
0	3000	3000	3000	2886	2740
10	3000	3000	3000	2857	2704
20	3000	3000	3000	2845	2684
30	3000	3000	3000	2842	2700
40	3000	3000	3000	2846	2693
50	3000	3000	3000	2844	2694

The over torque increase is controlled so that there is rate limit on how fast the motor reacts to these sudden changes. The rate of increase is controlled with a rate limiter block so that the torque can only increase certain percentage per second. Decrease is also controlled similarly although it is allowed to be ten times larger.

It seems that allowing over torque reduces the allowable power but this power drop doesn't really seem to be effected with the amount of allowed over torque. The key factor seems to be that how fast the over torque can or is allowed to react to sudden changes. Faster rate of increase in torque means that the motor can react faster and therefore keeping the output torque higher at the moment of ice impact. This higher baseline in addition with the ice load means higher maximum loads and lower power. The power drop is however notably smaller than in the inertia results.

Over torque also reduces the variation in rotational speed. This is shown in Figure 16 with the comparison of cases where no over torque is allowed and with 50% over torque and 100 %/s rate of increase. The results are for ice class 1A super and the lower gear results are presented as it is the limiting factor for all cases.

Without over torque, the speed drop in propeller shaft is 75.5 %. This is quite significant and there seems to also be resonance excitation on load case 1. With 100 %/s rate of increase in torque the speed drop is reduced to 48.3 % drop from the nominal speed and the resonance effect is much smaller. With 50 %/s rate of increase the speed drop is 53.6 % which tells that even with lower acting motors the effects are beneficial considering the lower speed drop. The amount of speed drop can be controlled with the amount of allowed over torque.

The resonance frequencies are of course dependent on the general dynamics of the system and the way how this is reduced in this example doesn’t really work in all the cases. But if we consider that designer usually wants to avoid situations where there are nominal frequencies near the systems nominal operating conditions. The over torque should help because it keeps the speed closer to nominal range and therefore it should prevent resonant excitation.

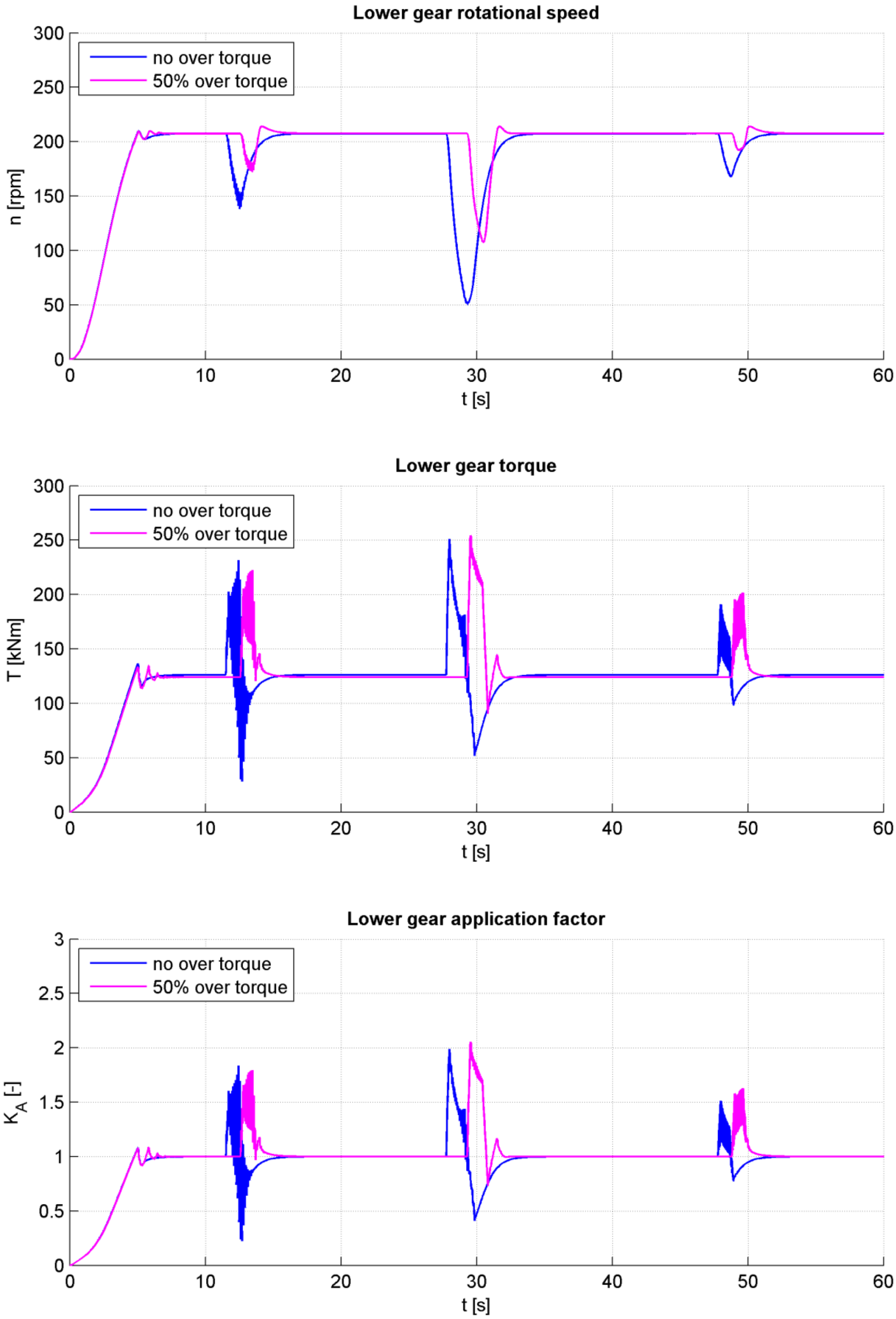


Figure 16. *Propeller unit speed drop with electric motor over torque.*

Although the influence on the resulting power is quite insignificant with the over torque, it should still be taken into account in the preliminary calculations because it improves the thrust performance (Riska 2011, p. 14-15). The influence on the speed is also significant because with a lower speed drop the resonant frequencies are not excited as easily. Cooling and controls just need to be designed accordingly so that the motor doesn't overheat. The most significant thing to take into account once again is the gear design itself, even though inertia has some significance like in the diesel engines. Cumulatively inertia may however have cumulatively bigger effect with other influencing characteristics.

6.4 System input speed

As was determined with both diesel engines and electric motors, the most restricting factor seems to be the gears. The gears depend on the nominal speeds of the power units and different gear pairs are needed for different system input speeds so that the optimal propeller rotational speed can be provided. In this US 305 case there are six different input speeds and all of these have their own gear sets.

Different gear ratios are also needed if the propeller diameter is changed. If we choose a smaller diameter propeller than the nominal, the propeller tip speed decreases. If the change in diameter is big, it may mean that the propeller tip speed is not on an optimal range anymore and the optimal range could be achieved again by choosing a different gear ratio.

The gears used in this study are designed for bollard pull and open water free running situations so there may also be so be some design issues when these are applied to the ice conditions. Usually in ice conditions scuffing and subsurface fatigue are the main types of failure and the safety factors against these conditions are the most demanding. This means that the lubrication and surface hardening respectively are the key factors to consider in the gear design.

Next we study how the systems input speed influences the allowed power. Both, engine and motor inertias are kept the same which is of course a bit inaccurate. Inertias vary with each rotational speed but this way the only varying factors are the gears itself. It also should be relatively easy to see which gears seem to perform better than the others. The results can be seen in Table 13 and those are iterated with 1 kW accuracy.

Table 13. *Results for the effect of input speed.*

		Iterated results [kW]			
Input speed [rpm]	**Nominal power [kW]**	**ICE-1C**	**ICE-1B**	**ICE-1A**	**ICE-1A super**
Diesel engine					
750	3000	3000	3000	2716*	2652*
800	3000	3000	3000	2650*	2590*
900	3000	2748	2644	2120	1988*
1000	3000	3000	3000	2550	2484
1200	3000	3000	2975	2384	2270
1600	3000	3000	2850	2354	2269
Electric motor					
750	3000	3000	3000	3000	2974*
800	3000	3000	3000	3000	2878
900	3000	2904	2859	2643	2529*
1000	3000	3000	3000	2951	2822
1200	3000	3000	3000	2868	2711
1600	3000	3000	3000	2661	2457

*) Engine stalls or the rpm drops below 40 % of the nominal

For diesel engines the power is limited with 750-1200 rpm gears by the upper gears and with the 1600 rpm lower gear. Most of these cases are limited by the subsurface fatigue safety factor but the 900 and 1200 rpm 1A super ice classes are limited by scuffing.

With electric motor the limiting loading condition is the subsurface fatigue in all of the inspected input speeds. Upper gear is limiting in the 900 and 1600 cases and lower gear is restricting the allowed power for the rest.

Scuffing seems to be more problematic type of load because it is only calculated with the maximum loading condition and if that calculation gives much lower safety factor that the rules require, the unit's power needs to be reduced relatively much more than in the case of subsurface fatigue. This can be especially be seen in the 900 rpm gears as the diesel engine simulation gives much lower allowed power even though the calculated excitation torque is quite close to the same as in the other cases. The small difference occurs because of the system characteristics.

There seems to be a trend that the higher rotational speeds have higher loads but because of the differences in the gear designs and the characteristics of the systems, we can't make too many conclusions how the input speed and gear ratio influences the loads on the gears. This is studied by making a simulation with nominal free running power and not making gear calculations, only the application factors for the gears are compared. The application factors can be seen in Table 14.

Table 14. *Application factors for ice classes 1A and 1A super with different input speeds.*

		Application factors			
Input speed [rpm]	**Nominal power [kW]**	**ICE-1A lower gear**	**ICE-1A upper gear**	**ICE-1A super lower gear**	**ICE-1A super upper gear**
Diesel engine					
750	3000	1.873	1.831	1.928	1.886
800	3000	1.919	1.886	1.958	1.922
900	3000	2.024	1.998	2.075	2.048
1000	3000	2.086	2.065	2.133	2.109
1200	3000	2.167	2.156	2.232	2.220
1600	3000	2.277	2.271	2.324	2.317
Electric motor					
750	3000	1.533	1.449	1.629	1.524
800	3000	1.553	1.476	1.650	1.552
900	3000	1.639	1.559	1.736	1.650
1000	3000	1.690	1.614	1.796	1.708
1200	3000	1.773	1.709	1.912	1.833
1600	3000	1.944	1.892	2.119	2.099

The load seems to rise as the input speed increases and the propeller speed is kept more or less the same in all cases. One reason for this may be that the higher gear ratio amplifies the effect of the inertia on the input shaft. This effect is however reduced as the faster rotating motors tend to have lower inertia as was seen with the electric motors earlier. However as some of the electric motors with different output speeds had close to same inertias, so this seems to be further contributing the power drop.

The application factors also show that the problem really is with the 900 rpm gear design because the load increases with higher gear ratios and input speeds. Still the allowed powers are lower than on any other gear pair in Table 13. This can be even seen with the ice classes 1C and 1B. This also shows that a good understanding about the gear design and loading conditions is needed in order to manufacture units that can withstand high ice loads.

Based on this finding it would be beneficial to place the power unit right on the propeller shaft without any reduction gears. This would also make the calculation much easier as the gear calculation wouldn't be an issue anymore. However this is not always possible for financial reasons so choosing a small inertia, low rotational speed engine seems to be the best case if the gear designs are good. Also some other component would restrict the design then which doesn't make this problem that much easier.

Possibly further studies could be done about making modifications to lower gear as well as the upper gear in order to find out if there is an optimal solution for the gear ratio when both gear sets are modified. Otherwise the only solution is to revise existing gear designs and to take into account that those may be used in the ice classified thrusters as well. Another way to lower the loads could be with better placement of flexible elements in the system.

6.5 Flexible coupling

Flexible coupling is a quite critical component in the marine propulsion power transmission system because it is basically the only component with any notable damping properties. These couplings are used to reduce vibration, noise and to protect system from misalignments and in most of the cases it has to be individually chosen for each project. It’s just that the nonlinear properties of the couplings that make these components very difficult to evaluate. The stiffness and damping values vary with the loading conditions and the operating temperature.

Flexible couplings have relatively low inertia compared to other parts of the power transmission so the stiffness and damping values are inspected. Nominal damping values however seem to be the relatively the same in elastic rubber ring element couplings even when the size changes. Therefore only the effect of stiffness is studied here. There are also other coupling types available so further studies are needed. It would be important to know if for example fluid couplings would be a better choice for ice classified thrusters.

Even though only the effect of stiffness is studied here, great care needs to be put in the choice of the flexible coupling because it has an effect on the system’s nominal frequencies. This can be seen as higher loads if the systems speed is on a critical level and this kind of a case is plausible because the ice loads cause rotational speed to drop. In Table 15 nominal frequency excitation can be seen in the electric motor case. With higher stiffness values the speed drop caused higher vibrations and therefore maximum allowed power couldn’t even be determined.

Table 15. *Results for the effect of flexible coupling stiffness.*

		Iterated results [kW]			
Stiffness [kNm/rad]	**Nominal power [kW]**	**ICE-1C**	**ICE-1B**	**ICE-1A**	**ICE-1A super**
Diesel engine					
277.01	3000	3000	3000	2708*	2646*
346.26	3000	3000	3000	2737*	2679*
415.51	3000	3000	3000	2745*	2686*
484.77	3000	3000	3000	2728*	2672*
(nominal) 554.02	3000	3000	3000	2716*	2652*
831.03	3000	3000	3000	2690*	2617*
1108.03	3000	3000	3000	2665*	2612*
Electric motor					
109.02	3000	3000	3000	2823	2666
136.28	3000	3000	3000	2832	2665
163.53	3000	3000	3000	2833	2678
190.79	3000	3000	3000	2864	2706
(nominal) 218.04	3000	3000	3000	2868	2711
261.65	3000	3000	3000	2873	2499
327.06	3000	3000	3000	2699	-

*) Engine stalls or the rpm drops below 40 % of the nominal

The diesel engine results show the same trend as do the findings of Barro and Lee (2014). Higher coupling stiffness seems to cause higher torque but interestingly the most optimal case seems to be around 70-75 % of the nominal range. With electric motor the results show the chosen flexible coupling may not be optimal because the results change so drastically with higher stiffness.

The optimal stiffness values for diesel engine can be explained by the fact that the nominal stiffness is given for cold coupling. Warm coupling values are nominal stiffness and damping values multiplied by 0.7 (Vulkan 2014, p. 24). The cold case is still however relevant as this kind of situation happens for example as the ship is leaving the port and the coupling has not yet warmed. It also seems to give higher vibratory torque which means that it has to be taken into account on the design and it determines power for sales offers. In this case the flexible coupling is quite optimal but there is a chance that the flexible coupling may also cause more oscillating torque if it is not chosen correctly.

With electric motor we can show that the flexible coupling is the most critical element of the system because it can basically determine how the system reacts to excitation. This has a great effect if the ice loads evoke speed drop to the unwanted range. In that case it would cause unnecessary power drop that could be avoided with a better flexible coupling choice. Resonant excitation is however harder to avoid in higher ice classes because of the higher loads and speed drops. Bigger speed drops are much more likely to excite natural frequencies. Using the flexible coupling data from an old project may be a good idea but if it is used with a different input speed, it needs to be remembered that it may cause some unwanted behavior.

Nominal frequencies are also a problematic thing if we consider the final design because the speed may drop to this unwanted range even if it is well below the normal operation range. The simulated case is the maximum ice load case which is thought to happen once during the propellers lifetime. If the speed drops lower than or stays above the resonance speed would occur, then the effect is minimal and it doesn't impact the design that much. When the speed drops almost exactly to the resonance range the loads are higher and it would affect to the flexible coupling choice. This doesn't make sense because the nominal running operation is much more critical condition than the high once-in-a-lifetime ice load. Also if the highest ice load doesn't excite the resonance frequencies some other load could. So balancing between the maximum ice load excitation and nominal running condition on the flexible coupling choice has to be made with great care. These kinds of cases are very problematic when considering the flexible coupling in addition to its dependency on the torque and input speed.

It seems that the coupling may be the most complicated of the components because of its nonlinear nature and its dependency on the application. It needs to be chosen based on the rotational speed and torque of the application but additionally the knowledge about the maximum transient condition is needed (Centalink 2008, p. 5; Vulkan 2014, p. 10-13). Engines and motors are much easier in a sense because their behavior is more predictable and only the total mass moments of inertia need to be taken into account in the preliminary calculations.

6.6 Propeller

There are several parameters that can change with each designed propeller but these variables can be most influenced by the designer when compared to the externally bought components. Customers also usually require that the vessel is designed for certain operational condition which limits the range of propeller design. But it is still good to know how much or what kind of effects the ice class related factors have, especially in the preliminary calculations.

Parameters needed for the calculation are hub and propeller diameter, pitch, rotational speed and ice class. FP and CP type propellers are considered to behave similarly although there are slight changes how the CP propellers pitch is taken into account in MCR condition. This doesn't have that much of an effect as the FP and CP propeller may still have similar pitch values. Another parameter that influences the final result is the propeller inertia used in the simulation.

Excitation torque determines how big the load is that influences the whole system. But because it is not determined by the systems power, it stays the same as long as the propeller design is the same. This is why the excitation torque is studied first with the parameters related to the excitation torque calculation. The other parameter that has effect on the power is the propeller inertia and it is studied later.

Propeller speed is studied first because sometimes, when choosing a smaller diameter propeller than nominal for the thruster unit, propeller speed needs to be increased in order to achieve the optimum blade tip speed. This affects excitation torque which is studied based on the equations (1)-(4). The results can be seen in Figure 17. All of the other parameters are kept constant (including D/d ratio). Open and ducted propellers with different diameters are studied separately.

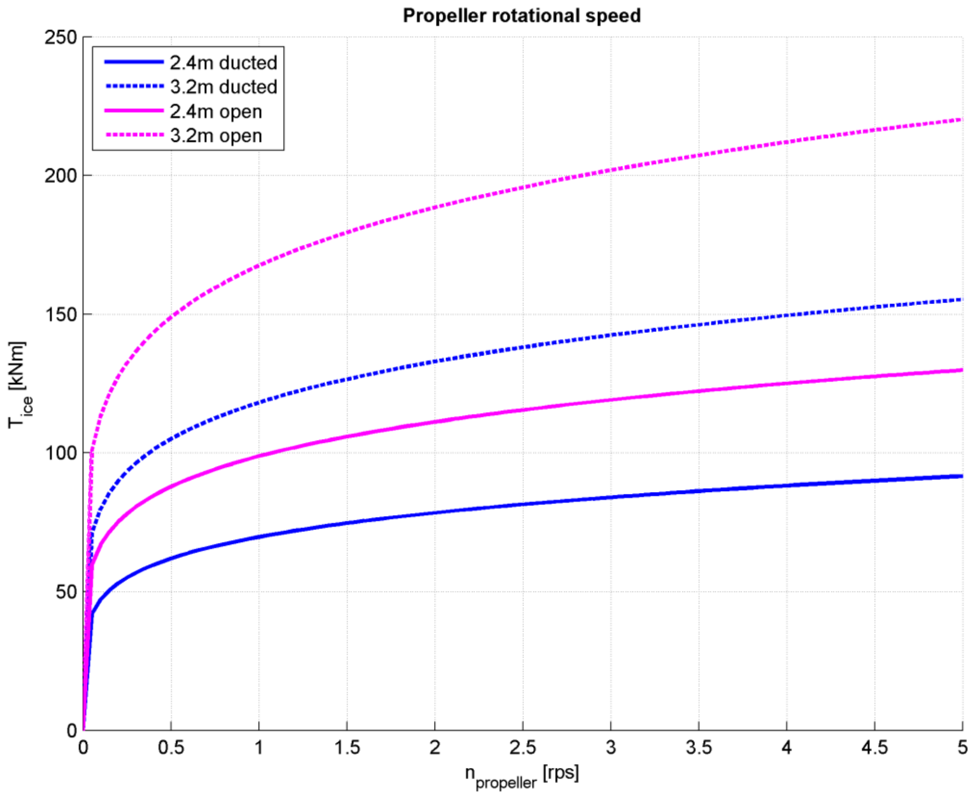


Figure 17. *Propeller speed (ICE-1C).*

Propeller diameter doesn't have an effect on the shape of the curve but it has an impact on the excitation torque magnitude. Higher diameter propellers have higher excitation torques. This seems to be the case almost always even if the smaller diameter propeller rotates faster. Additionally open propellers have notably higher torques so that the 2.4 m open propeller has close to similar torque level as the 3.2m ducted propeller. Otherwise rotational speed has relatively small effect in the MCR condition.

These effects are similar for propeller pitch since it is defined in a same manner in the equations and this is presented in Figure 18. Pitch can change in the propellers depending on the design goal for example. Open water and bollard pull propellers have different pitch values and other requirements and design choices also influence it.

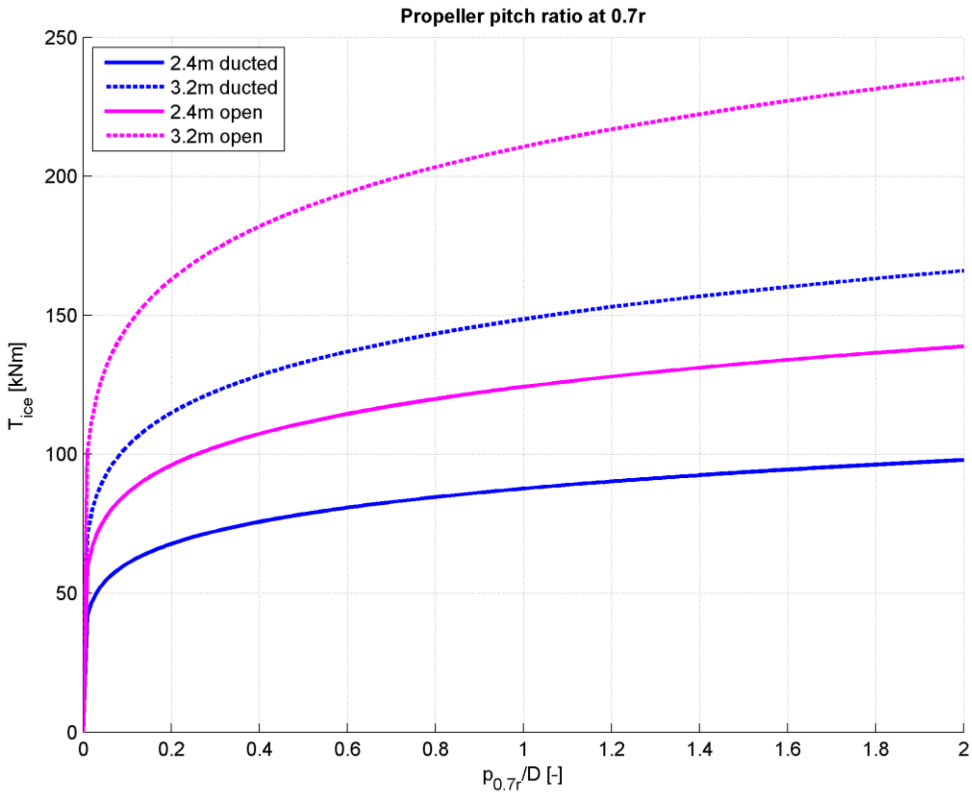


Figure 18. *Propeller pitch (ICE-1C).*

These figures were presented for ice class 1C but the effect is similar for higher ice classes as well. The difference in higher ice classes is that the loads are generally higher but the shape of the curve is still the same. Additionally there are higher requirements for propeller blade strengthening in higher ice classes and therefore the propeller blades are usually thicker.

In Figure 19 we can see how the ice class influences the maximum encountered loads on the propellers. It is calculated by using equations (1)-(4) and keeping all other parameters constant (including D/d ratio). Open and ducted propellers are also calculated separately to see what kind of effect it has on the maximum load.

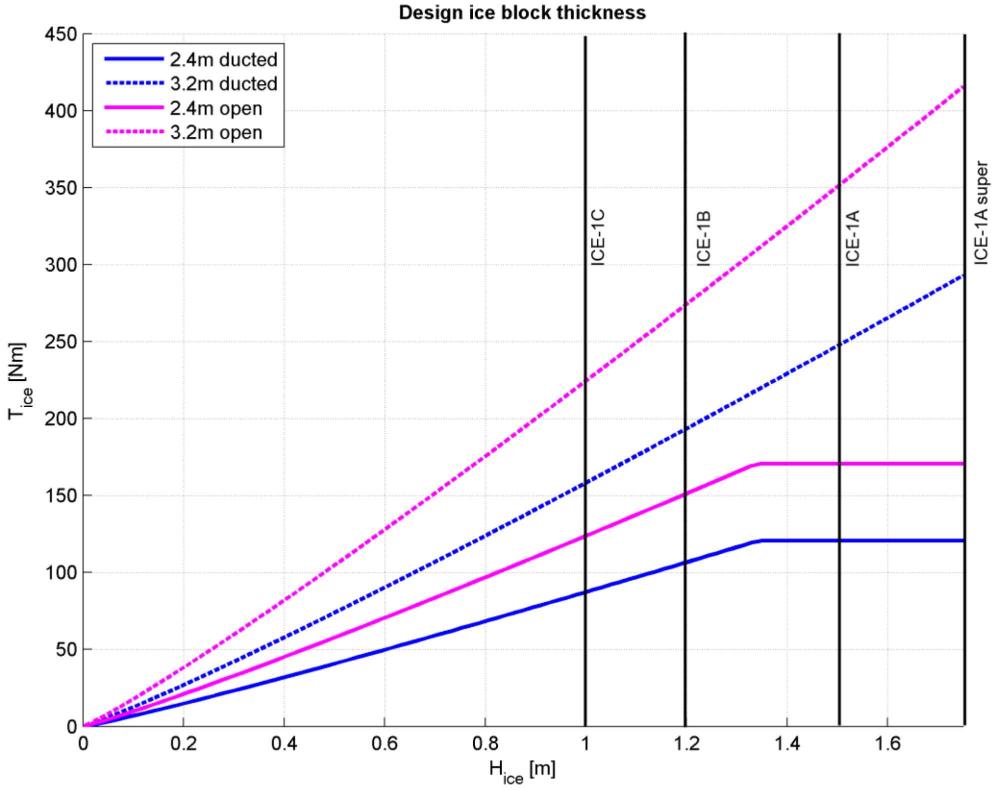


Figure 19. *Propeller ice torque.*

It can be seen that the ducted propellers (represented by blue lines) have a lot lower torque excitation. This also makes sense in a way that the nozzle protects from the biggest ice blocks but since the rules don't take any note on the prolonged effect of the nozzle blockage, that effect needs to be thought out by the propeller designer. Open propellers on the other hand are much more exposed to higher ice blocks and loads because there aren't that many structures that protect the propeller from them. Therefore the risk for blocking the water inflow for the propeller is also much smaller.

It is logical that the higher ice classes have higher loads but it seems that at the certain point there comes a limit to this. This can especially be seen with the 2.4 m propellers (represented by solid lines) as opposed to 3.2 m propellers that grow linearly. This is explained by the increase in ice block size and inertia. The geometry between blade and ice interaction becomes more asymmetric and the ice block tends to rotate away from the contact (Norhamo et al. 2009). This limits the maximum value for excitation torque in the rules and it depends on the propeller diameter.

This limiting diameter used in equations (1)-(4) can be calculated from the equation D_{limit}=1.8H_{ice}. These limiting diameters are listed in Table 16 for each ice class and they are the same for both open and ducted propellers.

Table 16. *Propeller torque limiting diameter for each ice class.*

Ice class	Design ice thickness H_{ice} [m]	Propeller diameter limit D_{limit} [m]
ICE-1A super	1.75	3.15
ICE-1A	1.5	2.7
ICE-1B	1.2	2.16
ICE-1C	1.0	1.8

This means that propeller diameters below the limit won't have higher excitation torque when choosing higher ice class. For example 2.7 m propeller has the same maximum ice excitation torque on ice classes 1A and 1A super. Therefore there is a significant drop on the excitation torque between 2.8 m and 2.7 m propeller in ice class 1A super. So choosing a slightly smaller diameter propeller can be beneficial if it can provide the desired thrust. This is at least the case if a designed propeller diameter is close to the limiting value. The number of load cycles will still change between ice classes however and this means that propeller unit is influenced by the ice class. This kind of limit can be seen in polar classes as well because of the formulation.

In most cases propeller speed is kept relatively the same and the propeller pitch can be found on a certain range which limits the effect of some design parameters quite much. Diameter is a significant factor in the design but at certain point that doesn't have as big of an impact on the excitation as the choice between open and ducted propeller. Open propellers will always have about 42% higher excitation torque if the diameter is the same. That means higher maximum torques for the other components which usually leads to lower allowed power.

Furthermore diameter and other design factors have great impact on to the propeller inertia which affects the systems dynamics. In ice classes the propeller blades also need to be strengthened because of the acting ice forces. This means that the blades are thicker than in the open water propellers and usually this results in a higher expanded blade area ratio and higher inertia.

The effect of inertia in water is studied by choosing a propeller designed for tug use without ice class. The propeller data is shown in Table 6. Keeping the other parameters same, inertia is then varied and the resulting power levels are presented in Table 17. The inertia varies a lot depending on the ice class, propeller design and the intended usage. Therefore the inertia is varied to be either higher or lower when compared to the nominal open water propeller so that the effect could be seen better generally.

Table 17. *Results for the effect of propeller inertia.*

Propeller inertia [kgm^2]	Nominal power [kW]	Iterated results [kW] ICE-1C	ICE-1B	ICE-1A	ICE-1A super
Diesel engine, open CP propeller					
1376.8	3000	3000	3000	2676*	2620*
1548.9	3000	3000	3000	2693*	2629*
(nominal) 1721.0	3000	3000	3000	2716*	2652*
1893.1	3000	3000	3000	2733*	2672*
2065.2	3000	3000	3000	2750*	2692*
2237.3	3000	3000	3000	2767*	2712*
2409.4	3000	3000	3000	2785*	2728*
Electric motor, ducted FP propeller					
1904.0	3000	3000	3000	2830	2624
2142.0	3000	3000	3000	2853	2697
(nominal) 2380.0	3000	3000	3000	2868	2711
2618.0	3000	3000	3000	2874	2724
2856.0	3000	3000	3000	2888	2737
3094.0	3000	3000	3000	2908	2752
3332.0	3000	3000	3000	2911	2770

*) Engine stalls or the rpm drops below 40 % of the nominal

Although these results may not be applicable directly because of the differences in the propeller designs, these give a general idea how the propeller inertia influences the load levels. It seems that the higher inertia propellers would dampen and resist the vibrations much more than the lower inertia propellers. One reason is that higher propeller inertia resists more the changes in rotation. This kind of effect somewhat happens also naturally as the higher ice class propeller blades need to be strengthened and therefore have higher inertia. Similarly inertia also increases as the diameter increases so the higher excitation loads are compensated a little bit.

Based on this, open water propeller designs are a good starting point for sales offers and preliminary calculations as these give a little bit higher loads and lower powers. This gives some additional safety for the design phase and there are much more open propeller designs available from the previous projects for a propeller manufacturer. From the design point of view, higher inertia is beneficial considering the load levels, but the propeller also is slower to accelerate so a balance point needs to be found.

While comparing the results for ducted and open propellers, it would seem that the ducted propellers are much better because of the lower excitation torques and therefore a bigger diameter propeller or higher power can be used. However, there is a possibility that flow to the propeller may be blocked because ice blocks get stuck on the nozzle. This could cause some unwanted behavior but on the other hand nozzle protects the blades from huge ice blocks and blade tip loads (Koskinen et al. 1996, p. 21). The risk of blockage is notably higher if the propeller diameter and depth are small. Therefore some consideration is needed which type of a propeller is better and how well it fits with the vessels operations. Ducted CP propeller with diesel engine would be the best design choice for the Arctic operations (Lee 2008, p. 20-21) but it should be noted that in severe ice conditions CP-mechanism will need strengthening which may make the design more complicated.

6.7 Best versus worst case scenarios compared to DIN standard results

The ice class calculation process is very demanding when viewed from sales perspective. First the correct thruster needs to be chosen from the unit selection and then calculations are needed to find out the final allowed power for the unit. This is especially hard because of all the uncertainties and unknown factors that need be taken into account. And it is even harder with the application dependent and nonlinear components like flexible couplings.

Luckily old projects and sold units can be used as a starting point for the ice classified units. Open water designs at least for propellers should be on the safe side and basic prime mover data is somewhat easily available on the manufacturer’s websites. This of course means a lot of work for the sales unlike in the open water cases where it is easier to make some kinds configurators which in turn make the sales much easier. In ice classes it’s not that easy because there are much more nonlinearities and variables in the designs. Also there are much more little details that affect the final results. It also requires a lot of co-operation with component suppliers so that the best possible result could be achieved.

The choice of the power unit determines quite many characteristics of the system because of the different output performances. Electric motors can provide full torque on a much wider range than diesel engines and electric motors can furthermore be allowed to have over torque. The most defining factor is however the motors rotational output speed or systems input speed which determines the used gears. The best and the worst performing engines and motors are listed on Table 18.

Table 18. *Comparison of best and worst case for diesel and electric motor.*

		Iterated results [kW]			
Power unit	**Maximum continuous rating**	**ICE-1C**	**ICE-1B**	**ICE-1A**	**ICE-1A super**
		Best case			
MAK 9M25C (435 kgm^2)	3000 kW, 750 rpm	3000	3000	2716*	2652*
WEG MGF 8808 (126.5 kgm^2)	3150 kW, 1000 rpm	3000	3000	2955	2825
		Worst case			
W9L26 (183 kgm^2)	2925 kW, 900 rpm	2823	2748	2397*	2344*
WEG MGF 12007 (405.9 kgm^2)	3150 kW, 900 rpm	2771	2685	2292	1850

*) Engine stalls or the rpm drops below 40 % of the nominal

Generally the electric motors can handle much higher loads and can therefore be equipped with bigger propellers but the biggest effect is on the system dynamics. Diesel engines are much more prone to rotational speed drop during ice milling which is very disadvantageous because the output performance is greatly affected. In the worst case this means that the engine possibly stalls.

The biggest difference comes from the different rotational speeds and gear designs related to those. Just because the gears are not optimized for ice conditions severely reduces the allowed power with ice loads and this can be seen even on the lower ice classes in the worst cases. Revising the gears so that every rotational speed in the unit could achieve similar powers would be the best solution and it would ease the sales quite much. Other factors can also have cumulatively some effect but these can be taken into account with the amount of load bins and possibly some small safety factor for the sales value.

The safety factor however needs to be considered quite carefully because it can take into account much of the uncertainties and variables but it can’t be too high. If the safety factor would be too high, then the thruster units are not as competitive anymore. It doesn’t seem so essential in the cases where the limiting factor is subsurface fatigue for gears but scuffing is much trickier.

This process proves to be much more challenging with other gear standards and rules because some of those don’t at least yet consider the subsurface fatigue. Also because the load spectrums are taken into account in different ways it means that there are considerable differences in the results. In Table 19 the results based on DIN 3991 gear calculation with ISO 6336-6 load spectrum are calculated for the best and the worst cases of DNV gear calculation results.

Table 19. Comparison of best and worst case to DIN 3991 based results.

		Iterated results [kW]			
Power unit	**Maximum continuous rating**	**ICE-1C**	**ICE-1B**	**ICE-1A**	**ICE-1A super**
Best case					
MAK 9M25C (435 kgm^2)	3000 kW, 750 rpm	3000	3000	3000*	3000*
WEG MGF 8808 (126.5 kgm^2)	3150 kW, 1000 rpm	3000	3000	3000	3000
Worst case					
W9L26 (183 kgm^2)	2925 kW, 900 rpm	2925	2925	2925*	2925*
WEG MGF 12007 (405.9 kgm^2)	3150 kW, 900 rpm	3000	3000	3000	3000

*) Engine stalls or the rpm drops below 40 % of the nominal

The iteration is done against American Bureau of Shipping classification society requirements for safety factors of gears and there are remarkable differences. Even though the gear geometries and lubrication properties stay the same, DIN gear calculation gives much higher results for the allowed power. This can partly be explained by the fact that the DNV scuffing calculation is much more sophisticated and takes more different small things into consideration. DNV also takes into account subsurface fatigue which is not considered in the older DIN standard at all. Nevertheless both of these marine gear rating means are widely used.

It also has to be remembered that this study is thought from the point of view of maximizing the allowed output power for the thruster but there are many other factors that the customer also requires or is interested in. Reliability, maintainability and cost are also big factors that influence the customer's final decision. The results in here are just ways to improve currently available products furthermore.

7. FUTURE DEVELOPMENT

The ice class rules are still under constant development and there is supposed to be updates in nearby future for both the Finnish-Swedish ice class rules and Polar class rules. These updated rules try to harmonize current rules better so that the basis for the design would be similar internationally (Kinnunen et al. 2013, p. 4) and the updates also try to better describe some design conditions that are not in the current rulings. A lot of these changes for Finnish-Swedish ice class rules were discussed in the meeting between Rolls-Royce and VTT in Rauma (2015).

Some updates are also considered for the ship structures but these seem to be mainly revises of the descriptions. The major changes that have impact on thruster manufacturers and influence the area of the study are more of an interest here. For example one of the major changes is the fact that there will one additional ice load milling case that has to be taken into account in the ice load simulation. This will have an effect on the simulation model that was made during this study as this case has to be added into the propeller block.

Probably one of the biggest changes is that there will be a possibility to make the ice class calculation also in the frequency domain (Trafi 2015a, p. 42-44). This might also be an interesting choice for sales offer calculations because the frequency domain is supposed to be more conservative. The difference is that the blade order excitations are described as continuous half sine waves as opposed to time domains transient half sine waves. (Trafi 2015a, p. 39) It could be worthwhile to study if this could be adapted to sales offer calculations easier and how much results differ from each other.

Although this might be a good way for estimation of the power for sales offers, there is still the problem that it requires the mass elastic data for the whole system to be designed beforehand. This is one of the biggest problems in the current rulings because the vessels have different operational requirements which results in different kind of components and mass elastic data for the propulsion units. So it would be better if the amount of design before the sales could be reduced and the system optimization would be done in the actual design phase.

The rules will also include load cases for ice ridge penetration as it is common that in the Sea of Botnica ice ridges form on the sides of the ice channels. The ridges form because the navigating ships push small ice blocks below the sides of the channel and those can slowly form formidable ice ridges as the winter progresses. Ships then have to penetrate these ridges as they have to maneuver around the oncoming vessels for example.

This ridge penetration may then cause forces on the thruster body especially if the ships back in to them thruster first. These forces may be concentrated on the thruster hub, rear end, nozzle or the thruster body forming a moment arm and therefore causing increased loads on the steering gear and hull fitting. This kind of situation is at least considered very likely for the 1A super ice classed ships which are supposed to sail independently on difficult ice conditions but it is also relevant for lower ice class ships. (Trafi 2015a, p. 53-55) This kind of load can also be caused by an ice block so this kind of scenario seems quite relevant in the ship design.

The design ice block can cause similar loads on thruster hub, body or rear end as well as nozzle upon impact. The impact can cause loads on the axial and lateral direction. Currently the Finnish-Swedish rules don’t really take any note on this and the future update will give a design loads for these kind of situations. (Kinnunen et al. 2013, p. 5)

A completely new requirement is also proposed for the azimuthing thruster body which can also experience significant amount of vibration in ice conditions. This amendment would make it necessary to estimate the thruster body natural frequencies so that they wouldn’t be excited by the propeller order resonance frequencies. (Trafi 2015b, p. 26-28) This is of course meant to lower the risk for damages in the thruster components but this also introduces some other uncertain features that have to be taken into account, like the thruster’s structural damping in water and the stiffness of the hull attachment. The most important vibration modes to be considered in the rules are longitudinal and transverse but vertical vibration may have some importance as well.

All of these loading conditions are still in development and under review so anything certain can’t really be said how the future rules will affect the thruster design. Some updates are although needed in the current ice classification rules generally as some rules don’t take into account all the possible solutions available today. Also a simplified way to evaluate propeller loads for sales would be welcomed.

8. CONCLUSIONS

The phenomenon of the propeller-ice interaction is very complex and challenging. This can also be seen in the ice class rules as they require simulation for the ice classified units for assessing the maximum component loads. This means that basically a complete design is needed also in the preliminary calculations for the sales so that the sales offer values can be estimated more accurately. This makes selling thrusters really complicated because the simulation requires so much information which is usually uncertain when sales offers are made.

In this study calculation tool for sales department usage was developed which includes the whole process for the Finnish-Swedish ice classes and gear calculation according to the DNV Classification notes 41.2 (2012) and DIN 3991 (1998) gear rules. The goal is to determine maximum allowed power for the chosen azimuthing thruster unit so that every propulsion machinery component can be classified. The bevel gears are considered the most restricting component because heavy torque loads have a big impact on the torque transmission capability. Sensitivity analysis was also conducted so that the effect of different component choices could be better understood and better approximation for sales offers could be chosen.

The program was made with Matlab and it contains the calculation of maximum encountered torque loads and simulation with the described excitation cases according to the ice class rules. Maximum loads are evaluated with the simulation model and these are then used in the gear calculation. Gears are evaluated because those are the most power restricting components on the power train. The final power value is then iterated by lowering the propeller unit's power and therefore lowering the experienced total maximum torque load. This is done so that the gears meet at least the minimum required safety factors and so that it would be possible to design this kind of a thruster at the later stages.

The developed program can be applied for every propeller unit by changing the gear data, unit's mass-elastic data and modifying simulation model accordingly. It can also be applied into the L shafting configuration but the some work needs to be done in the main program that calls for the calculation functions. In both cases some expertise is needed to modify the simulation model in order to describe project unit and configuration but once the model is made, it can be used in the evaluation as long as the rules and gear data are kept up-to-date and the unit's mass elastic model is considered a valid reference.

The program can also be utilized as a classification tool but more work needs to be done on the output data. All the necessary calculation results are available but those are not saved into excel sheets or printed in any other way so the output files need to be created. Also some approvals may be necessary from the classification society.

In the upcoming rules there is a possibility to alternatively make the same kind of analysis in a frequency domain instead of the time domain analysis done in this study. This may be more easily approached solution as the current way to do this is a quite complex for the sales offers. The frequency domain analysis is also supposed to be more conservative which may be helpful as the resulting power could possibly be used in the sales offer without additional consideration for safety. Although the analysis still requires the mass-elastic system data which is the most problematic feature of the rules currently.

The uncertainties in the mass elastic data were studied with sensitivity analysis and the developed program. Rolls-Royce US 305 azimuthing thruster unit was used as a basis for the analysis and the inspected properties were power unit inertia, electric motor over torque, system input speed, flexible coupling stiffness and propeller design parameters and inertia. Propeller design on one hand can be affected the most by the designer but on the other hand the intended usage profile sets some limitations. It means that the designer needs to be aware how the choices affect the final design. Power unit and flexible coupling are more critical in a way because those can be externally bought components. The propeller unit designer can't influence their properties as much. Power unit also defines the final systems dynamic characteristics and system input speed which has an impact on the choices for other components.

Based on the sensitivity analysis gear design seems to be the most critical thing. The gears studied were foremost designed for the open water conditions and their geometry, lubrication and other properties are optimized for that. If the gears are not designed to work in the heavy loading conditions like ice loads some gears seem to have problems related to safety. Subsurface fatigue can be taken into account in sales offers with the accuracy of the load spectrum used in the gear calculation but scuffing is much more restricting. Scuffing seems to cause significant power drop and it can only be affected by either lowering the nominal power or lowering the excitation torque. Contact stresses or tooth root stresses did not restrict the final allowed power in any case.

Some gears could probably use some redesigning if they're going to be used in ice conditions so that the allowed power would be similar for each system input speed. It would ease the sales process so that every gear pair and input speed doesn't need to be calculated independently.

These notions were made with the DNV Classification Notes 41.2 gear rules and the results were compared to DIN 3991 gear calculation results. DIN rules gave much higher power ratings because those are not as sophisticated on the scuffing calculation and the DIN rules don't take into account the subsurface fatigue. Another difference is in the load spectrum which is based on the equivalent damage and the gears are evaluated against infinite life with the equivalent torque.

Flexible coupling also seems to be quite critical component in some cases because of the systems natural frequencies and excitation damping. Because the ice excitation torques cause speed drop, it may excite the natural frequencies of the system. This would cause higher loads and is therefore one of the key things to look for in the simulation results. If the resonance excitation happens, the resonance frequency can be changed by choosing a different flexible coupling. Because the coupling's properties also depend on the surroundings and loading conditions it is one of most complex components to choose. Using a coupling from a previous project as a reference may be a good approximation but it needs to be remembered that it may cause problems in certain circumstances.

Another tricky feature in the simulation is the power source inertia because it's heavily related to the engine's output speed and structure. Different output speeds have different inertias and even on the same output speed the inertias vary very much because of the different amounts of cylinders or number of poles. The total mass moment of inertia also affects the power so that the lower inertia allows more power. So choosing the lowest possible inertia in the power source seems to be beneficial and electric motors are favorable because of the wider range of full output torque and possibly better overall efficiency. Diesel engine solutions are on the other hand often cheaper solutions therefore those are used more often.

Electric motor over torque is a little bit disadvantageous for the power but it significantly reduces the speed drop. The most noticeable thing seems to be how fast the motor can react to sudden changes in the system. Faster reaction reduces the speed drop but it also lowers the allowed power more. Over torque is however beneficial because it reduces the variation in thrust but it also causes the electric motor to heat up. Overheating must be prevented in the design.

Propellers are mostly designed according to customer's demand which means that the propeller is either optimized for bollard pull condition or free running condition. Therefore there are usually certain ranges that the design parameters are and their effect is quite minimal in the final propeller loads. The most significant things in the rules are propeller diameter and the choice between the open and ducted propeller. Open propellers have much higher loads than ducted propellers but other aspects may limit the usage of ducted propeller, like nozzle blockage. Higher propeller inertia seems also to be favorable in the final design but open water propeller designs are a good starting point for preliminary sales offer calculations.

There is a big difference between the best possible allowed output power and the worst possible output power in one thruster unit. This is emphasized even more with the different approaches in the gear rules. The goal is of course to design the best possible unit that can be made but worse choices need to be taken into account in the preliminary calculations and sales offers. Some additional safety could be used for more conservative sales offers so that the promised power can be achieved and the risk is minimized.

Safety factor for the power is however financially delicate question, small safety assures better competitive characteristics, higher takes into account more inaccuracies in the preliminary calculations. Ultimately it's up to the designer how well he or she can optimize the system and how much there is time available to do this. However, this safety factor could be assumed smaller if the buyer can give out more detailed data for the engine or vessels operational profile before the actual offer is made. It would also reduce the amount of guessing that needs to be done.

It has to be remembered that this is a very limited view on the thruster design for the ice classes and many other things also affect it. Vessels operational profile and available space may also affect certain decisions which add requirements to the thruster design. Customers also desire reliability, maintainability, good performance, energy efficiency and low cost so the final design is much more complicated than just optimizing propeller and gear design for the maximum output power.

With heavy ice conditions some other means of power transmission could also be better. For example hydrostatic transmission could be plausible as it is used in some heavy machinery like excavators and earth-moving equipment. The benefit would be that the propeller could be driven directly with a hydraulic motor like in the podded drive solution but the difference being in the way how the torque loads act on the components. The torque peaks that happen can be restricted with a pressure relief valve so no other machinery component would be affected by the ice loads that the propeller experiences. This of course would need expertise in the hydraulic design and probably even a new design for the azimuthing propulsion unit. But it could be worthwhile in at least some very specialized purposes.

Some other ways of minimizing the ice loads could for example be alternative choices for shafts, like quill drives. Those would allow more radial flexibility than the current system's steel shafts and flexible coupling. Also the placement of the flexible coupling in the shaft line could have some significance on the component loads.

Optimal navigation through ice fields would be still best solution for minimizing the ice loads and it would also help minimizing fuel consumption and travel time (Kotovirta et al. 2008). This of course doesn't affect the current rules because there's still a possibility for navigating ships to experience high ice loads but the amount of encountered ice loads could be reduced.

New updates in the rules also bring some other things that need to be considered in the thrusters like the propeller order excitation effect on structure and ice block collisions with the thruster body. These kinds of additions will require more from the thruster structural design but those also should make the navigation safer. It’s a good thing that also other approach to this time domain simulation process is to be expected but it doesn’t seem to remove the biggest problem with the current rules. It would still require almost complete thruster design for the sales in order make the sales offer more safely.

REFERENCES

ABB, High voltage induction motors, Technical catalog for IEC motors, Sep. 2011, 152 p. Accessed: 22.04.2015 Available: http://www09.abb.com/global/scot/scot234.nsf/veritydisplay/593ffc2bb0b08efac1257917003fae1c/$file/HV_Induction_motors_technical_IEC_catalog_FINAL_EN_092011_lowres.pdf

R. D. Barro, D. C. Lee, Transient Torsional Vibration Analysis of Ice-class Propulsion Shafting System Driven by Electric Motor, Transactions of the Korean Society for Noise and Vibration Engineering, Vol. 24, No. 9, pp. 667-674.

R. Browne, L. Norhamo, Background Notes to Propeller Ice Interaction Loads, IACS Unified Requirements for Polar Ships, Rev. 2, 30 Jun. 2007, 15 p.

CAT, Torsional Vibration analysis MAK 9 M25C, CAT Report No. 12025049, 11 Dec. 2012, 40 p. Restricted access: Rolls-Royce Intranet

Caterpillar Motoren GmbH & Co. KG, M25 C propulsion project guide, Apr. 2012, 108 p. Accessed: 22.04.2015 Available: https://marine.cat.com/cda/files/3774080/7/Project%20Guide%20M25C%20Propulsion%2004.2012.pdf

Centa, Centalink the silent driveshaft catalog, Jul. 2008, 15 p. Accessd: 22.04.2015 Available: http://www.centa.info/data/products/43/int/cl-07-08_01.pdf

Det Norske Veritas, Ice Strengthening of Propulsion Machinery, DNV Classification Notes No. 51.1, Jan. 2011, 63 p. Available: http://exchange.dnv.com/publishing/cn/2011-01/Cn51-1.pdf

Det Norske Veritas, Calculation of Gear Rating for Marine Transmissions, DNV Classification Notes No. 41.2, May 2012, 61 p. Available: http://exchange.dnv.com/publishing/cn/2012-05/CN41-2.pdf

Det Norske Veritas, Ships for Navigation in Ice, Rules for Classification of Ships Part 5 Chapter 1, Jul. 2014, 162 p. Available: http://exchange.dnv.com/publishing/rulesship/2014-07/ts501.pdf

Trägfähigkeitsberechnung von Kegelrädern ohne Achsversetzung, Teil 1-4,DIN 3991, Sep. 1988, 52 p.

P. Eriksson, K. Riska, J. Vainio, Traffic restrictions to Finnish and Swedish ports, Winter navigation research board: Research report No 58, Helsinki Mar. 2007, 27 p. Available: http://www.trafi.fi/filebank/a/1352716465/6f7686f4619c32e342b06a69726d5e9f/10736-Report_No_58_Traffic_Restrictions_to_Finnish_and_Swedish.pdf

T. Feese, C. Hill, Prevention of torsional vibration problems in reciprocating machinery, Proceedings of the thirty-eighth turbomachinery symposium, Texas, Sep. 2009, p. 213-238

P. Häger, Bevel gears for thrusters, Dynamic positioning conference, Houston, USA, 18-19 Sep. 2011, 17 p

International Organization for Standardization, Calculation of load capacity of spur and helical gears – Part 6, ISO 6336-6, Switzerland, 15 Aug. 2006, 24 p.

A. Kinnunen, M. Tikanmäki, J. Heinonen, P. Koskinen, M. Jussila, Azimuthing thruster ice load calculation, VTT Research report No 87, Espoo 17 Feb. 2013, 40 p. Available: http://www.trafi.fi/filebank/a/1410168004/306ddce167e20359fd880b7ff0297373/15354-Report_No_87_Azimuthing_thruster_ice_load_calculation.pdf

P. Koskinen, M. Jussila, H. Soininen, VTT Research noters: Propeller ice load models, VTT offsetpaino Espoo, 1996, 82 p.

V. Kotovirta, R Jalonen, L. Axell, K. Riska, R. Berglund, A system for route optimization in ice-covered waters, Cold Regions Science and Technology, Volume 55, Issue 1, Jan. 2009, p. 52-62

E. J. Lecourt, P. B. Zahn, Considerations for propellers and propulsion plants operating in northern latitudes, U.S. Navy Symposium on Arctic/cold weather operations of surface ships, 3-4 Dec. 1985, pp. 175-182

S-K. Lee, Combining ice class rules with direct calculations for design of arctic LNG vessel propulsion, CasTech conference, Bangkok, Thailand 10-13 March 2008. 22 p.

G. Magazinović, Transient torsional vibration analysis of marine propulsion plants. 21. simpozij Teorija i praksa brodogradnje (In memoriam prof. Leopold Sorta), 2014, pp. 505-512.

MAN Diesel & Turbo, L27/38 Project Guide – Marine, Jun. 2014, 400 p. Accessed: 22.04.2015 Available: http://www.mandieselturbo.com/download/documents/ProjectGuides4Stroke/L27-38_GenSet_TierII.pdf

MAN Diesel & Turbo, L32/40 Project Guide – Marine, Nov. 2013, 278 p. Accessed: 22.04.2015 Available: http://www.mandieselturbo.com/download/documents/ProjectGuides4Stroke/L32-40_GenSet_TierII.pdf

MAN Diesel & Turbo, L35/44DF Project Guide – Marine, Aug. 2014, 413 p. Accessed: 22.04.2015 Available: http://www.mandieselturbo.com/projectguidesengine/data/L35-44DF%20IMO%20Tier%20II%20%E2%80%93%20Marine.pdf

Mathworks, Matlab Help: Choose a solver. Accessed: 09.04.2015 Available: http://se.mathworks.com/help/simulink/ug/choosing-a-solver.html#f11-41989

E. J. Nestorides, A Handbook on Torsional Vibration, Cambridge University Press, England, 1958, 663 p.

L. Norhamo, G. M. Bakken, O. Deinboll, J. J. Iseskär, Challenges Related to Propulsor – Ice Interaction in Arctic Waters, First International Symposium on Marine Propulsors, Trondheim Norway, Jun. 2009, 10 p.

J. Pollari, Voimansiirtolinjaston jääkuormitusten simulointi Simscapella, Rolls-Royce Oy Ab Rauma, 2 Apr. 2012, 22 p. Restricted access: Rolls-Royce Intranet

K. Riska, Design of Ice Breaking Ships, In Cold regions science and marine technology, United Nations Educational, Scientific and Cultural Organization (UNESCO) and Encyclopedia of Life Support Systems (EOLSS), 2011, 43 p. Accessed: 04.03.2015, Available: http://www.arctis-search.com/Design+of+Ice+Breaking+Ships

K. Riska, J. Kämäräinen, A review of ice loading and evolution of the Finnish-Swedish ice class rules, Proceedings of the SNAME annual meeting and expo, Houston, Texas, USA 16-18 Nov. 2011, 44 p.

W. Tong, Mechanical Design of Electric Motors, CRC Press, 2014, 736 p.

Trafi, Maritime Safety Regulation: Finnish-Swedish Ice Class Rules 2010, Transport Safety Agency Helsinki, 23 Nov. 2010, 48 p. Available: http://www.trafi.fi/en/maritime/ice_classes_of_ships

Trafi, Guidelines for the Application of the Finnish-Swedish Ice Class Rules, Trafi Helsinki, 20 Dec. 2011, 37 p. Available: http://www.trafi.fi/en/maritime/ice_classes_of_ships

Trafi, A new version of Finnish-Swedish Ice Class Rules 2010, Draft, Transport Safety Agency Helsinki, Jun. 26, 2015a, 60 p.

Trafi, A new version of the Guidelines for the Application of the Finnish-Swedish Ice Class Rules, Draft, Transport Safety Agency Helsinki, Jun. 26 2015b, 40 p.

VTT - Rolls-Royce meeting, P. Koskinen, A. Kinnunen, A. Sánchez-Caja, R. Vanninen, J. Keski-Rahkonen, Rauma 8 Apr. 2015

Vulkan Drive Tech, Technical Data, Highly Flexible Couplings for Industrial Applications, Jan. 2014, 128 p. Accessed: 22.04.2015 Available: http://www.vulkan.com/en-us/drivetech/Documents/DriveTech-Catalogues/VDT_HFC_TechnicalData_WEB_2014.pdf

WEG Electric Motor Selection, M-Line MGF Low and High voltage motors Accessed: 06.05.2015 Available: http://ecatalog.weg.net/TEC_CAT/tech_motor_sel_web.asp

Wärtsilä Ship Power Technology, Wärtsilä 26 Project Guide, Vaasa, Nov. 2013, 160 p. Accessed: 22.04.2015 Available: http://www.wartsila.com/docs/default-source/product-files/engines-generating-sets/medium-speed-engines/wartsila-o-e-w-26-pg.pdf

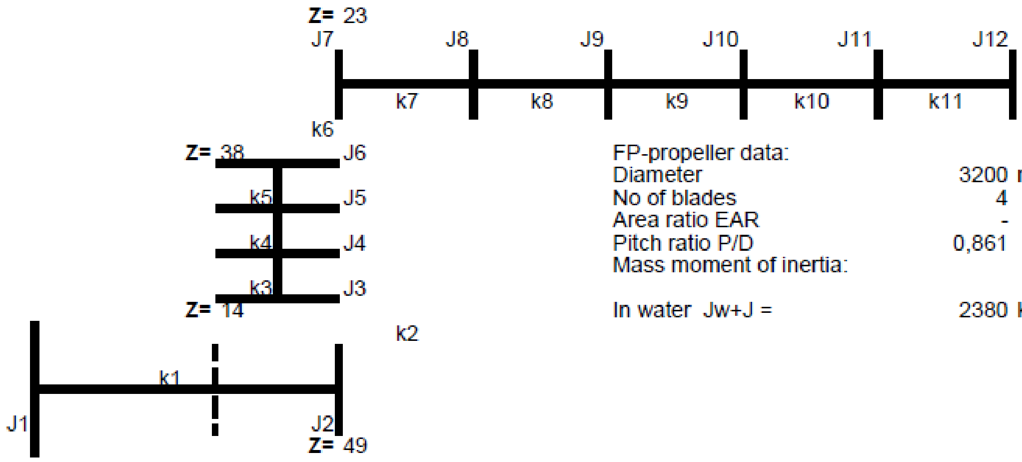
Wärtsilä Ship Power Technology, Wärtsilä 32 Project Guide, Vaasa, Jun. 2014, 204 p. Accessed: 22.04.2015 Available: http://www.wartsila.com/docs/default-source/product-files/engines-generating-sets/medium-speed-engines/wartsila-o-e-w-32-pg.pdf

Wärtsilä Ship Power Technology, Wärtsilä 34DF Project Guide, Vaasa, Feb. 2015, 204 p. Accessed: 22.04.2015 Available: http://www.wartsila.com/docs/default-source/default-document-library/34df-product-guide.pdf

APPENDIX 1: MASS ELASTIC DIAGRAM OF ELECTRIC MOTOR DRIVEN US 305

TORSIONAL VIBRATION SCHEME AQM US 305 FP P40 /4100

Output	3000	kW
Speed	1200	rpm
Torque	23,87	kNm



FP-propeller data:

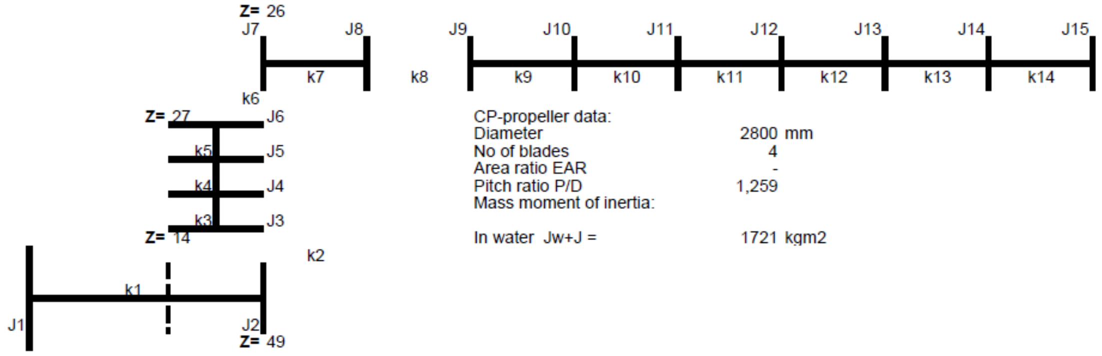
Diameter	3200 mm
No of blades	4
Area ratio EAR	-
Pitch ratio P/D	0,861
Mass moment of inertia:	
In water Jw+J =	2380 kgm2

	Description
1	Propeller
2	Lower wheel
1	Propeller
2	Lower wheel
3	Lower pinion
4	Gear coupling
5	Gear coupling
6	Upper wheel
7	Upper pinion
8	Flange
9	Flexible coupling
10	Flexible coupling
11	Shaft
12	Rotor
13	
14	
15	
16	
17	

APPENDIX 2: MASS ELASTIC DIAGRAM OF DIESEL ENGINE DRIVEN US 305

MASS ELASTIC DIAGRAM AQM US305CP /3970

Output	2666	kW
Speed	750	rpm
Torque	33,94	kNm

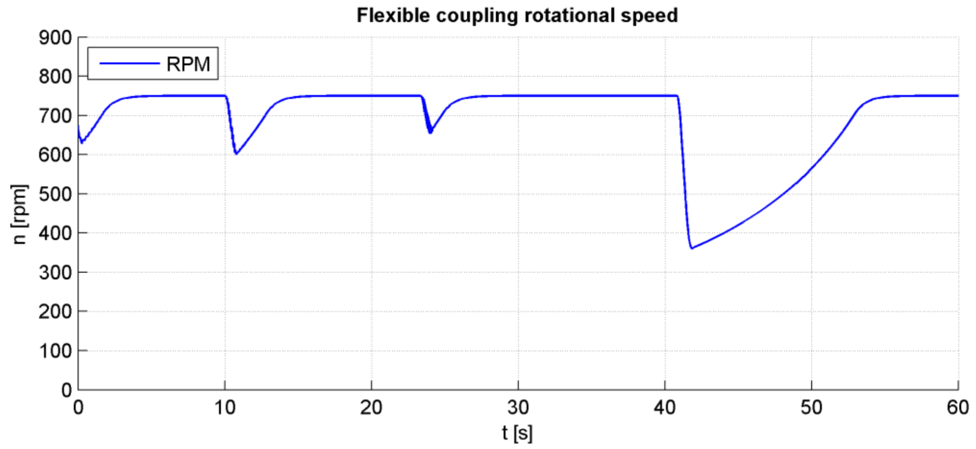


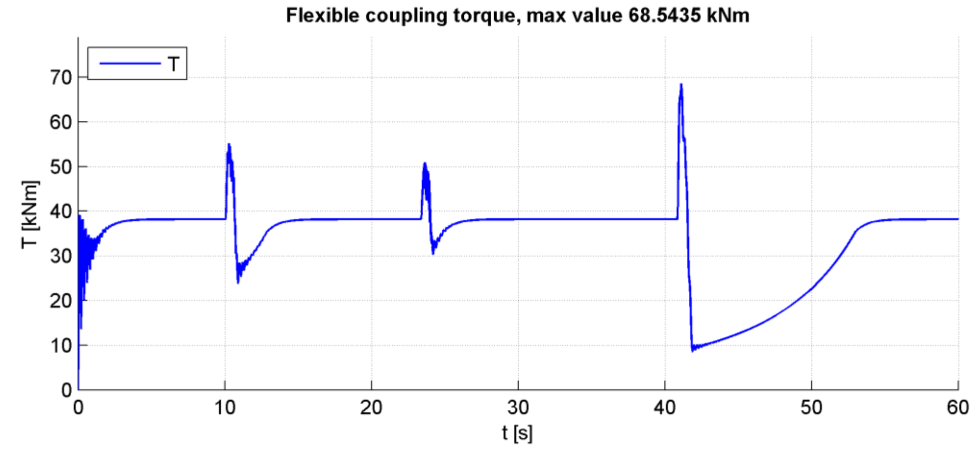
CP-propeller data:

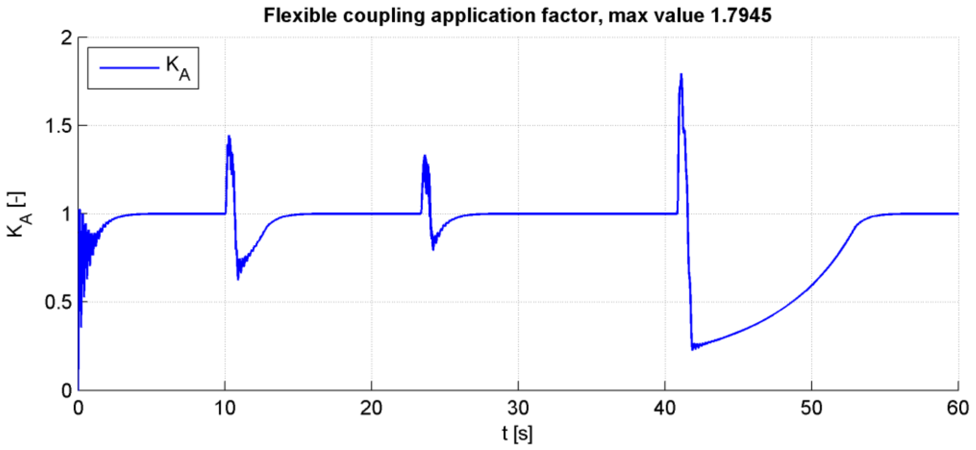
Diameter	2800	mm
No of blades	4	
Area ratio EAR	-	
Pitch ratio P/D	1,259	
Mass moment of inertia:		
In water Jw+J =	1721	kgm2

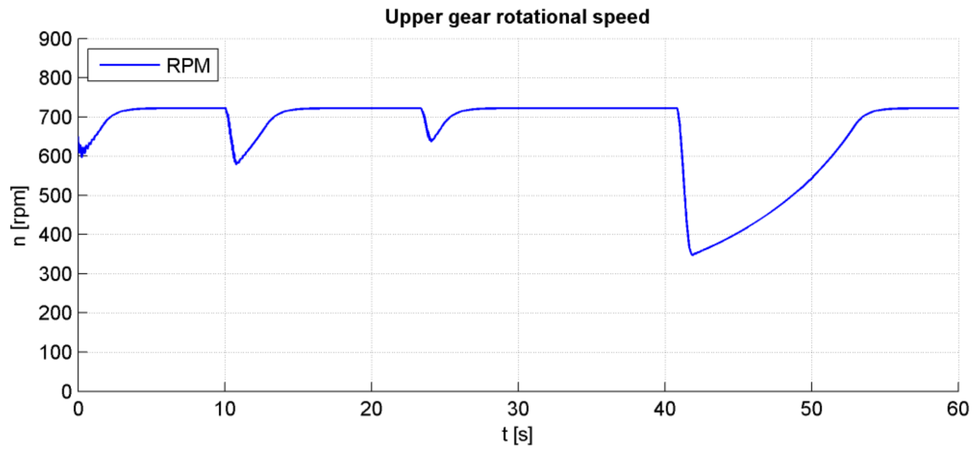
	Description
1	Propeller
2	Lower wheel
1	Propeller
2	Lower wheel
3	Lower pinion
4	Gear coupling
5	Gear coupling
6	Upper wheel
7	Upper pinion
8	Clutch
9	Clutch
10	Flange
11	Gear coupling
12	Gear coupling
13	Flange
14	Flange
15	Shaft
16	
17	

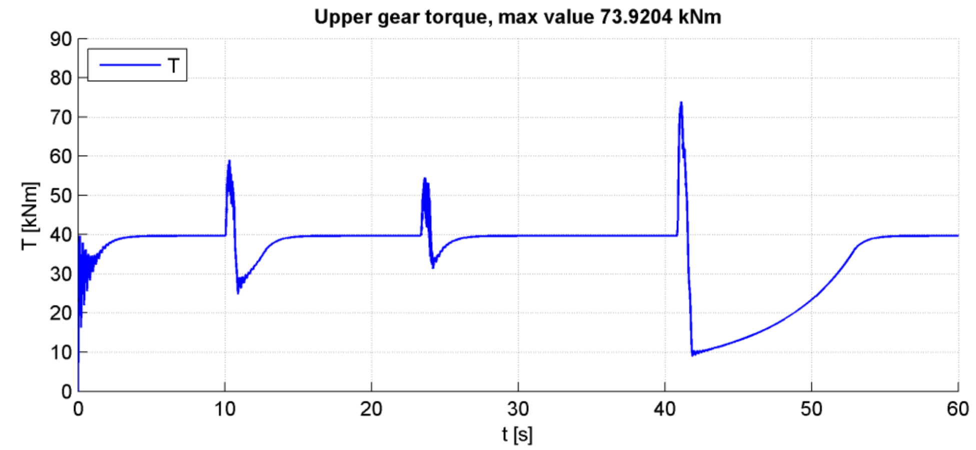
APPENDIX 3: EXAMPLE OF SIMULATION OUTPUT FIGURES

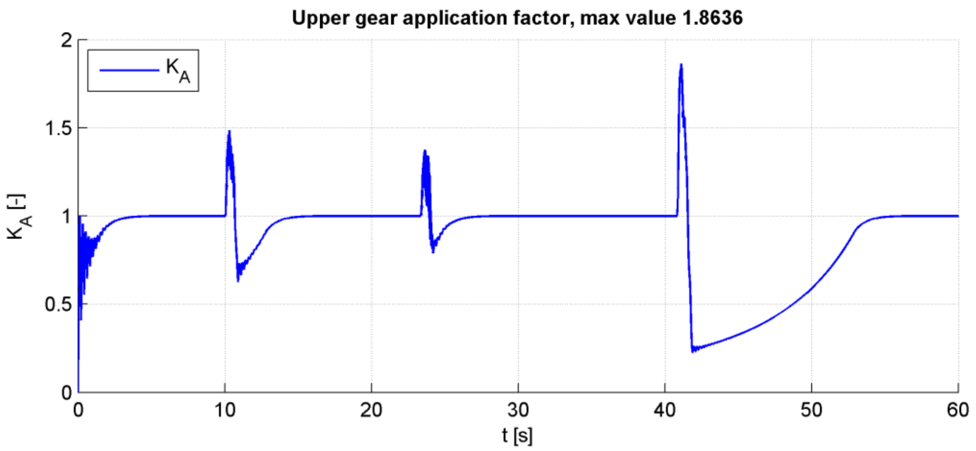


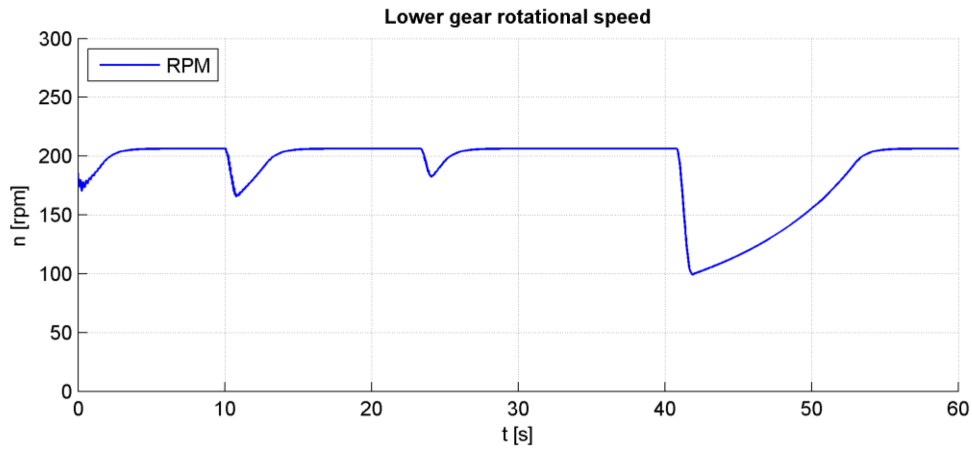


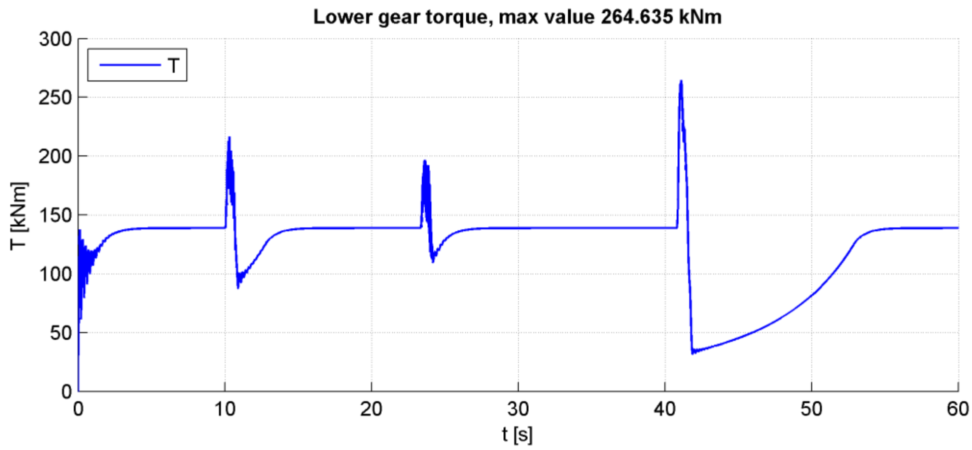


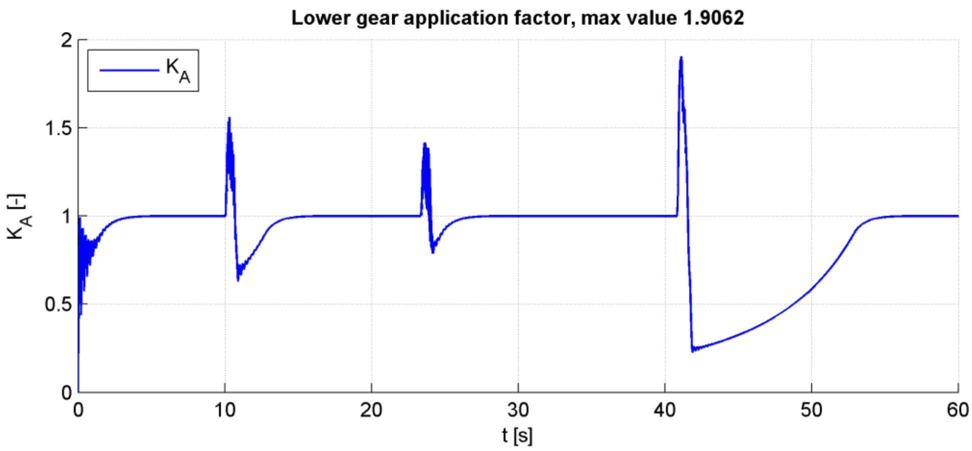

Upper gear rotational speed
RPM
n [rpm]
t [s]
900
800
700
600
500
400
300
200
100
0
0
10
20
30
40
50
60


Upper gear torque, max value 73.9204 kNm
T
T [kNm]
t [s]
90
80
70
60
50
40
30
20
10
0
0
10
20
30
40
50
60


Upper gear application factor, max value 1.8636
K_A
K_A [-]
t [s]
2
1.5
1
0.5
0
0
10
20
30
40
50
60


Lower gear rotational speed
RPM
n [rpm]
t [s]


Lower gear torque, max value 264.635 kNm
T
T [kNm]
t [s]


Lower gear application factor, max value 1.9062
K_A
K_A [-]
t [s]